STEAM TURBINES

AND

TURBO-COMPRESSORS:

THEIR DESIGN AND CONSTRUCTION

—BY—

FRANK FOSTER, M.Sc.

With over 240 illustrations, many of which are original, and have been made expressly for this work

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PREFACE.

There is a need, the author believes, for a book on Steam Turbines which shall not be simply a popular description of existing machines, nor yet a mathematical treatise, and which shall not be a mere compilation from the patent records.

In fact, an attempt has been made to produce a book which shall not be so deeply involved in mathematics, or so insistent on the necessity for the attainment of the highest thermal efficiencies, regardless of the thousand-and-one modifying practical influences, as to be at once above and beneath the intelligence of the practical engineer. It is hoped, indeed, that the treatment of the subject in this book is practical in the best sense.

A theory, to be true, should be a complete and rational explanation of *all* phenomena which come within its scope. There is no true theory in this wide sense, for the simple reason that our knowledge is not extensive enough to construct one. Consequently, we require to be very careful not to be too dogmatic in applying our necessarily imperfect knowledge to the manifold uses of practical engineering.

In particular, we must keep in mind the fact that it is the all-round commercial efficiency, and not the thermal or mechanical efficiencies, which decide the ultimate success or failure of a machine. It is to be hoped that this economic bias in the book will not frighten students from making use of it. In the author's opinion students can and ought to be trained to understand the influence of

economic considerations on the design of machinery. The lack of such training has, in the past, done a great deal to reduce the value of a technical education.

Most of the subject matter in this book is original. In particular may be mentioned two new heat diagrams—one for steam and one for gases—which will be found very useful in turbine design: a general method of determining the critical speed for any kind of a rotor; and a method of determining the best vacuum in the condenser of either a turbine or a reciprocator, under any given conditions as to load factor and the price of coal. In the chapter on diagrams and calculations the method of determining the principal quantities for turbines is explained and illustrated by examples.

In view of the widespread adoption of steam turbines for the propulsion of ships, their application in this direction has been made the subject of a special chapter. Gas turbines and turbo-compressors are briefly discussed, and for the sake of completeness an outline of turbine history has also been included.

The author here desires to acknowledge the assistance of various turbine builders in supplying data and illustrations, of Mr. E. D. Foster for kindly checking some of the calculations, and of Prof. Reynolds for permission to publish an account of what was probably the first multiple-stage reaction turbine put into operation.

In general, the source of information taken from the works of various writers is acknowledged in the body of the book.

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turbine might be infinite, yet unless the steam possessed the requisite velocity the turbine would not act. On the other hand, if the steam possessed the necessary velocity then the pressure might be—if that were practicable—absolutely zero, and yet the turbine would work quite normally.

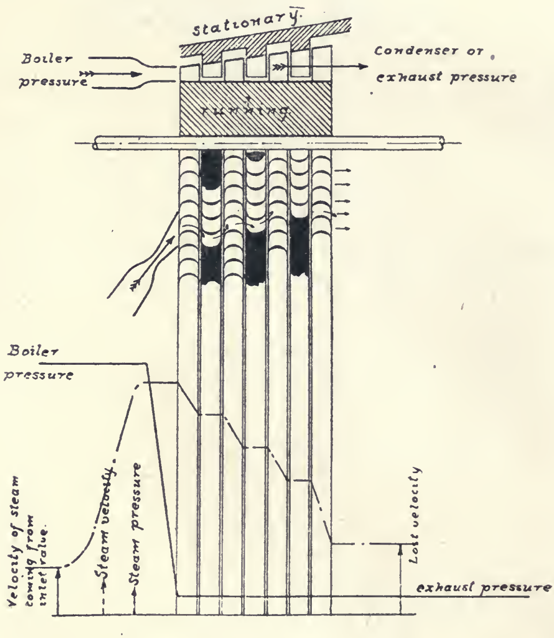

FIG. 2.—CURTIS TURBINE, ORIGINAL.

The statical steam pressure, even though varying from stop valve to exhaust, can cause no rotation, but it can produce end thrust on the blades. It is sometimes asserted that the fall of pressure towards the exhaust end of a Parsons turbine causes rotation, but the assertion is not correct.

To illustrate this, imagine a vertical turbine immersed in a tank of water, or, better still, in mercury. Then clearly there will be a continuous fall in the fluid pressure from the lower end of the turbine upwards. If only this fall of pressure would cause the turbine to rotate, what an inexhaustible source of cheap power we should have! A

few such turbines attached to the under-water portion of a ship would propel it regardless of winds or fuel. Our locomotives and our automobiles would consist mainly of a tank of water or mercury containing a vertical turbine geared to the driving wheels. Our factories and our power stations would be operated in the same fashion. Smoke would be banished, and our dense London fogs would be relegated to the debates of learned societies.

Indirectly the fall in pressure in a steam turbine does cause rotation. The fall in pressure generates velocity

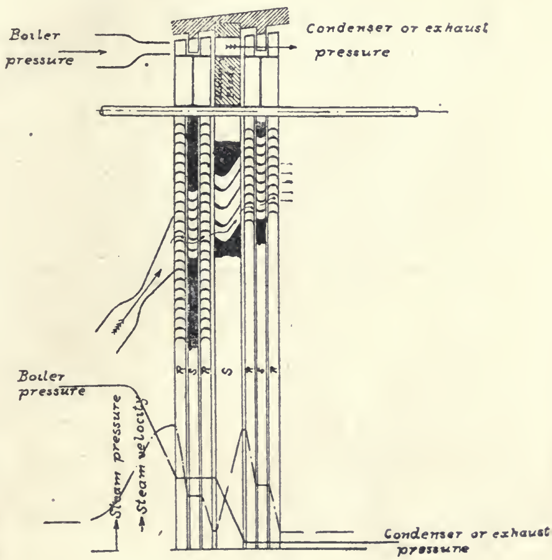

FIG. 3.—CURTIS TURBINE.

in the steam, and this velocity causes the turbine to rotate.

As the steam passes through the turbine the direction and amount of its motion is altered. In order to do this the blades must exert a pressure on the steam. It is this pressure, or rather the corresponding reaction of the steam on the vanes, which causes the rotation.

When steam expands adiabatically the amount of work which it does depends only on the initial state of the steam and its final pressure. In the ordinary steam

engine (neglecting losses) this work is done on a loaded piston. In the turbine it is done on the steam and shows itself as increased kinetic energy imparted to the steam. We shall have occasion to prove this later on, when we

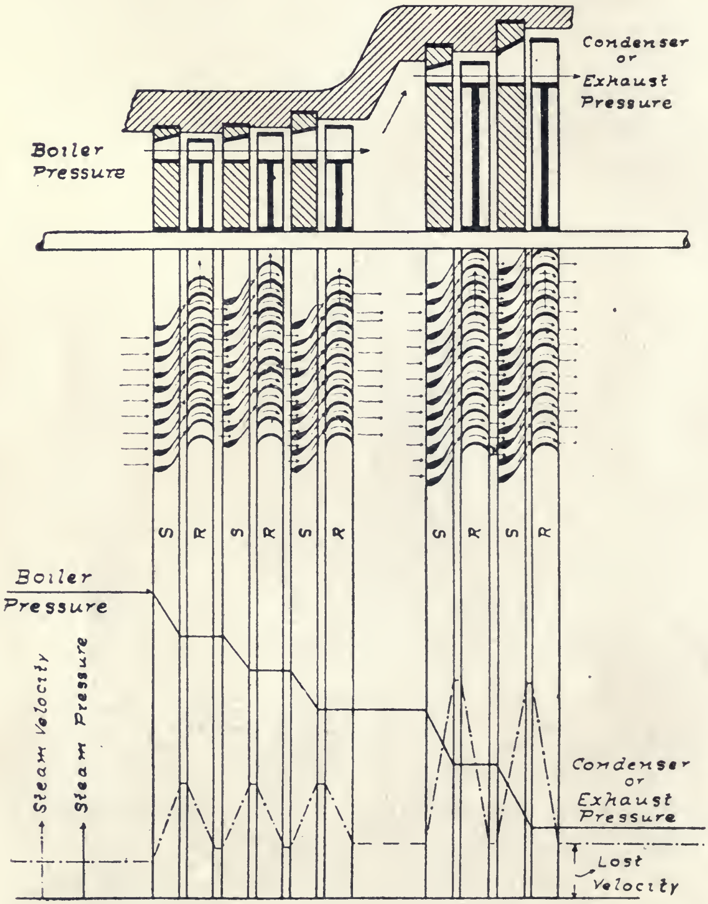

FIG. 4.—HAMILTON-HOLZWARTH AND RATEAU TURBINES.

come to deal with steam nozzles. In practice part of this kinetic energy is used up in overcoming frictional (including eddy) losses and re-appears as heat. Hence, the area of an ordinary indicator or entropy diagram does not represent the increase of kinetic energy in the

steam. The necessary correction is, however, very easily made for an entropy diagram.

Definition of a Stage.—In steam turbines we have, from a thermodynamic point of view, two principal members, First there is the member which is the means of causing kinetic energy to be generated in the steam. In impulse turbines this member is frequently called a nozzle. Then there are the moving blades which convert the kinetic energy of the steam into work on the shaft. In reaction turbines the latter member also serves as a generator of kinetic energy. In each stage we have a certain amount of kinetic energy—in excess of that previously existing—imparted to the steam, and its practically complete conversion into either, work on the shaft, or heat through eddy and frictional loss. At the end of a stage the steam is practically in the same condition, so far as its kinetic energy is concerned, as it was at the commencement of the stage. The expansion or generation of kinetic energy may take place wholly in the nozzles, as in impulse turbines, or partly in nozzles and partly in the moving blades as in reaction turbines. The absorption of kinetic energy may take place in one set of moving blades, as in most turbines, or in two or more sets, as in the Curtis turbines. In any case, however, we do not have any appreciable generation of kinetic energy in nozzles between two sets of moving blades of the one stage.

Two Classes of Turbines.—Steam turbines may be separated into two broad divisions according to the manner in which the steam is expanded and the resultant kinetic energy absorbed by the moving blades.

In one class of turbine there is no fall of pressure in the moving blades and hence no generation of kinetic energy. This kinetic energy is wholly generated in the fixed blades or nozzles as they are frequently called. The De Laval, Rateau, Zoelly, Curtis, and Riedler-Stumpf turbines all belong to this class. They are commonly called action or impulse turbines.

In the other class of turbine, part of the generation of kinetic energy takes place in the moving blades, which are at the same time absorbing kinetic energy. In this way the steam velocities are kept down and friction reduced. There is, of course, a fall of pressure in the

moving as well as in the fixed blades. The Parsons is the best known example of this class. Such turbines are spoken of as reaction turbines. As a matter of fact this classification into impulse and reaction turbines is not strictly correct, seeing that all turbines work by reaction.

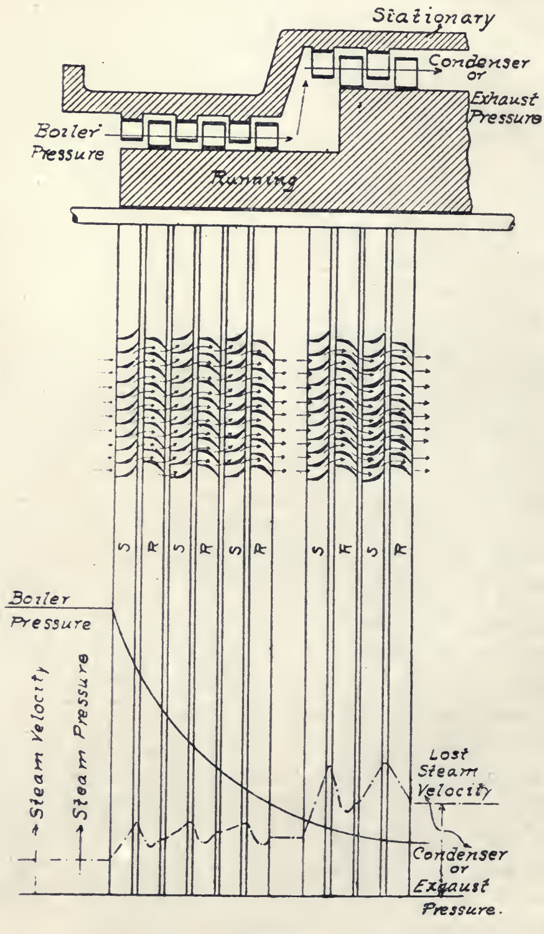

FIG. 5.—PARSONS TURBINE

In the De Laval and some forms of the Riedler-Stumpf turbine the whole drop in pressure takes place in one set of nozzles which direct the high velocity steam on to a single wheel. In the Rateau the fall in pressure is

divided into from 20 to 40 stages, each with its own set of nozzles and a wheel. In the Curtis the fall in pressure takes place in several stages (two, three, or four), but in each stage there are several sets of moving blades alternating with sets of fixed blades. The first set of fixed blades give the requisite drop in pressure and direct the high velocity steam on to the first wheel. The steam leaves this wheel with a high but much reduced velocity. It enters a set of fixed blades which re-direct it on to the second wheel. These intermediate fixed blades are merely guides and are not intended to act as generators of kinetic energy. A similar principle is adopted in some forms of the Riedler-Stumpf turbine.

In the Parsons turbine there are from 50 to 80 stages, each consisting of a set of fixed and a set of moving blades. There is a fall of pressure in every set of blades. Hence, kinetic energy is being generated continuously, and abstracted in every alternate set of blades.

We might design a turbine so that the whole of the kinetic energy is generated in the moving blades which also absorb it. In this case the velocity in the fixed blades would not change, and these blades would then be merely guides and not in any sense nozzles. So far, no attempt has been made to construct a turbine on this principle, although it is perfectly practicable.

Fig. 1 illustrates diagrammatically the distribution of pressure and velocity in a De Laval turbine. It will be seen that there is no fall in pressure after the steam has left the nozzle.

Figs. 2 and 3 illustrate the original Curtis turbine and the modern Curtis turbine so far as steam pressures and velocities are concerned. In the original design there was only one set of nozzles which discharged the steam at exhaust pressure and at high velocity into the moving and fixed blades, of which there were a large number of rows. In the modern design there are at least two and usually four complete stages in the expansion, each stage only having two, three, or four sets of moving blades with their corresponding guides. In each stage the pressure is approximately constant, but the velocity is

being continually reduced from the time the steam leaves the nozzles up to the time it enters the set of nozzles belonging to the next stage.

Fig. 4 illustrates the distribution of pressure and velocity in a Rateau or a Hamilton-Holzwarth turbine. The increase in the steam velocities at the change in the diameter of the wheels should be noted, as also the general form of the turbine blade sections.

In Fig. 5 we have illustrated the pressure and velocity distribution in a Parsons turbine. These five illustrations should help the reader to form a clear mental picture of the actions going on inside a steam turbine. The same general distribution of pressure and velocity would also occur in a gas turbine or a turbo air compressor. In the latter case the moving blades must not, however, be symmetrical, so that the De Laval, Rateau, and Curtis blade forms would have to be somewhat modified.

CHAPTER II.

DESCRIPTIONS OF TURBINES.

De Laval Turbine.—Fig. 6 gives a good idea of the main features of the steam end of this turbine. The single turbine wheel (or disc) *B* is mounted on a long, slender shaft *C*, and carries on its rim a continuous band of blades (otherwise called vanes or buckets) *A*. Steam from the boiler is expanded in the nozzles *D* (of which only four are shown in the figure) and leaves them with a velocity of from 2,500ft. to 4,200ft. per second, according

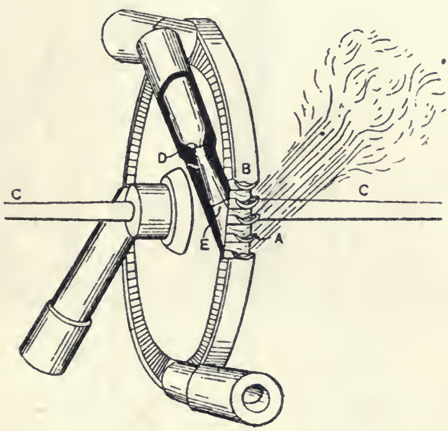

FIG. 6.—DE LAVAL STEAM TURBINE.

to the admission and exhaust conditions. Leaving the nozzles, the steam passes between the blades *A* and out to exhaust. The chief objection to this type of turbine is the high speed of the wheel which is necessary. For most efficient working the rim speed of the wheel should be nearly half that of the steam as it issues from the nozzle. The actual speeds employed are given in Table I.

TABLE I.—*Speeds of the Turbine Wheels.*

Size of Turbine.	Middle Diameter of Wheel.	Revolutions per Minute.	Peripheral Speed, Feet per Sec.
5 h.p.	100 mm., 4in.	30,000	515
15 ,,	150 ,, 6in.	24,000	617
30 ,,	225 ,, $8\frac{7}{8}$in.	20,000	774
50 ,,	300 ,, $11\frac{3}{4}$in.	16,400	846
100 ,,	500 ,, $19\frac{3}{4}$in.	13,000	1,115
300 ,,	760 ,, 30in.	10,600	1,378

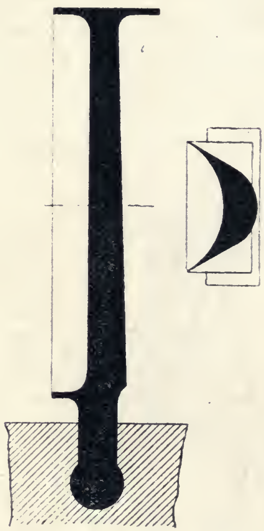

FIG. 7.—BLADE OF DE LAVAL TURBINE.

The cost of wheels for still higher speeds is said to be prohibitive.

With very few exceptions the turbine shaft drives one or two counter-shafts by means of double-helical gears, with a speed reduction of about 10 or 13 to 1. The gears have a linear velocity of from 100ft. to 150ft. per second. The pinion is in one piece with the shaft, and is of high-carbon crucible steel or nickel steel. The gears are of mild steel. Bronze gears which were once used were found to go brittle with age. These gears give a very smooth and remarkably efficient drive.

The bearings are all lined with white metal. Ring-oiling is used for the counter-shafts, but has not been found satisfactory for the high-speed wheel shaft. For the latter wick drips with sight glasses are used. The

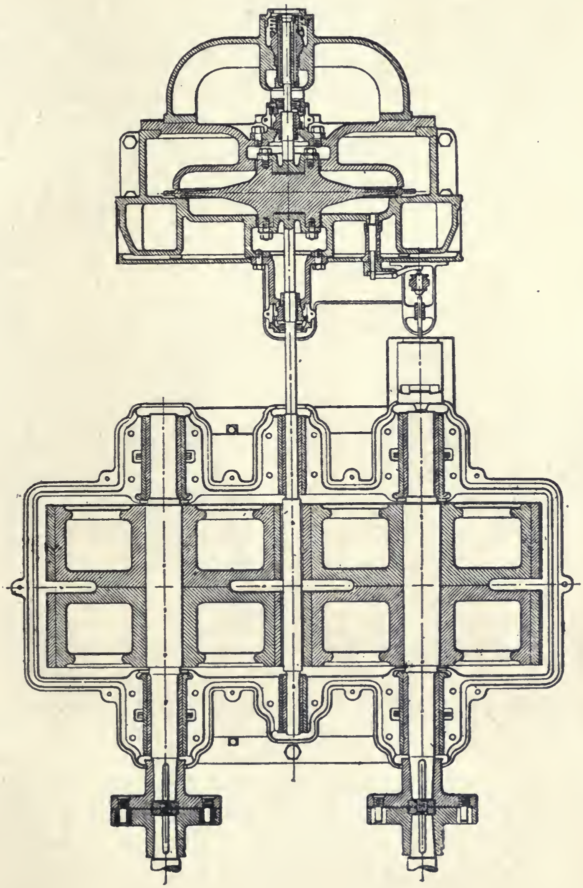

FIG. 8.—SECTIONAL PLAN OF DE LAVAL TURBINE AND REDUCING GEARS.

main bearings sometimes have a helical groove of very small pitch for distributing the oil.

For use with saturated steam, a steam separator is desirable. A gauze strainer to prevent the admission of foreign matter is used between the stop and throttle valves. Each nozzle is provided with a small shut-off

valve, thus enabling the number of nozzles to be varied according to the load, and making much throttling unnecessary. Throttling is a source of loss. As a rule the number of nozzles is only varied where—as in an electric lighting plant—considerable changes of load occur but seldom during the day, and then not without warning. Ordinary fluctuations such as are met with in power supply are provided for by the throttle governor. As a rule, the nozzles are of circular cross section, and the steam jet is cut by the plane of the wheel in a flat ellipse, the ends of which only partially fill their buckets, and thus lead to considerable eddy losses. To reduce

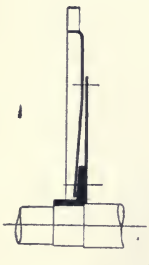

FIG. 9.
DISC OF RATEAU STEAM TURBINE.

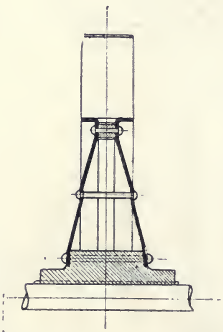

FIG. 10.
DISC OF HAMILTON-HOLZWARTH STEAM TURBINE.

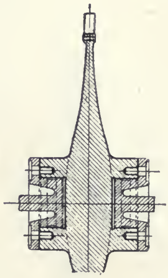

FIG. 11.
BUCKET WHEEL OF DE LAVAL TURBINE.

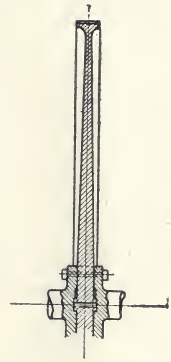

FIG. 12.
BUCKET WHEEL OF RIEDLER STUMPF TURBINE.

these losses the final section of the nozzle can be made rectangular, as in the Riedler-Stumpf turbine.

The high speed of a turbine wheel necessitates a special construction. The wheels of the larger turbines are solid throughout, the shaft being bolted to annular projections. In the smaller wheels there is a central hole for the shaft. The wheels are of nickel steel, considerably thickened up towards the centre to strengthen the wheel for resisting the great centrifugal forces it is subjected to.

The blades (Fig. 7) are of wrought steel, so shaped that when put in place they form their own shrouding. The angles of inlet and outlet are, according to Stodola, 30° and 31·5° respectively. The blade is thinned off a

trifle towards the outer end so as to diminish its centrifugal loading on the rim. Although these blades are expensive individually, yet there are not many of them, and they are easily replaced when necessary.

These turbines are sometimes belted to line shafting, belt speeds of 5,000ft. per minute being employed.

Fig. 8 illustrates the general arrangement of the turbine. It will be noticed that the gears constitute by far the largest portion of the machine. The factor of safety allowed in the turbine disc at the periphery where the stresses are greatest is about 5, so that the wheel would fracture (at the rim) at about double the normal

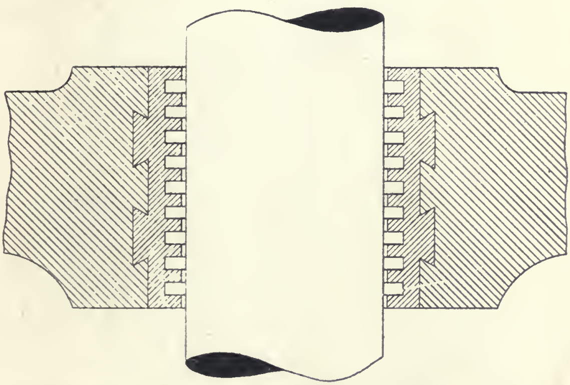

FIG. 13.—LABYRINTH PACKING FOR INTERIOR DIAPHRAGM OF AN IMPULSE TURBINE.

speed. The blades would fly off, and by considerably relieving the stresses on the rest of the disc reduce the chances of a serious accident.

Rateau Turbine.—This turbine, as we have seen, consists essentially of a series of De Laval turbines on the same shaft, each turbine delivering its exhaust to the succeeding one. The object of this arrangement is primarily to obtain a more manageable speed, and also to reduce the loss to the exhaust by reducing the final exit velocity of the steam. The rim speed of the vanes seldom exceeds 400ft. per second, and 350ft. per second, or even less, is a more usual figure. The number of stages depends upon the rim speeds and blade angles, but is usually between 15 and 40. The wheels are of sheet steel,

slightly conical, with a flanged rim to which the blades are riveted. In some of the later constructions a second

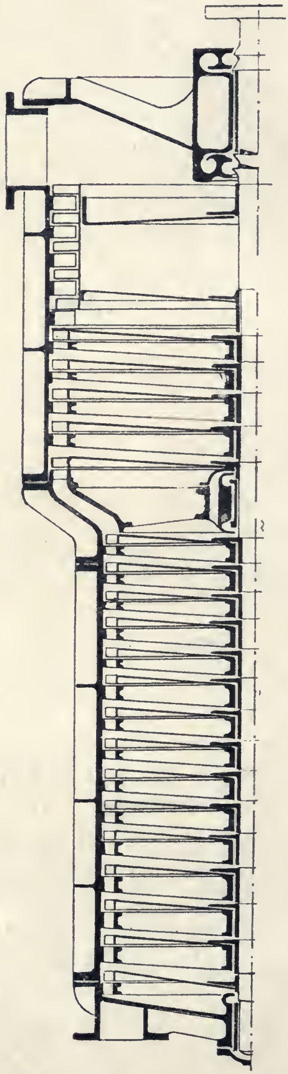

Fig. 14.—Section of Rateau Steam Turbine.

steel disc, forming a kind of base to the cone, is riveted to the main disc so as to prevent its straightening out under the influence of centrifugal force. (See Fig. 9.)

The fixed blades or nozzles are grouped together on a level with the moving blades, in fixed partitions or diaphragms. Thus between two consecutive wheels we have a diaphragm containing the nozzles (Fig. 14). These nozzles generate kinetic energy in the steam, and direct the high-velocity steam on to the moving blades, from which it passes to the next set of nozzles. Practically the whole of the expansion takes place in these nozzles,

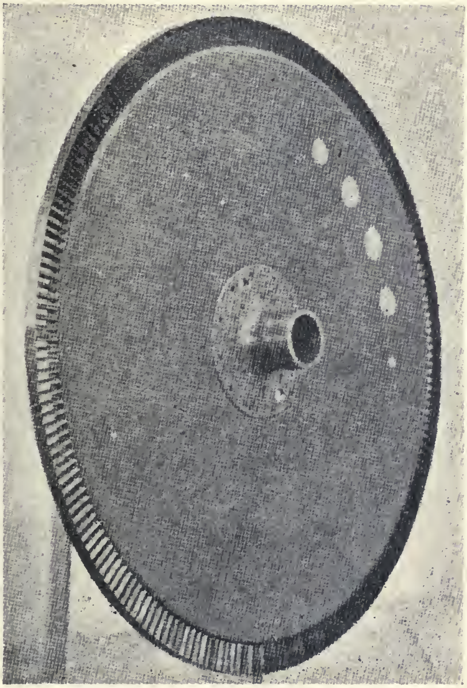

FIG. 15.—SINGLE WHEEL, RATEAU STEAM TURBINE.

there being no fall of pressure in the moving blades themselves.

Owing to the fall of pressure in each stage being small it is not necessary to make the nozzles diverging. Except in the very largest turbines, the nozzles only cover a portion of the circumference. To prevent excessive leakage from one stage to the next, the nozzle diaphragms extend from the casing to the shaft. A slight clearance between the shaft and the diaphragm is allowed, the

diaphragm having circular grooves cut in it (as indicated in Fig. 13) to prevent the leakage of steam. This kind of packing is the one most generally used in steam turbine work, a water or steam seal being added where there is a tendency for air to leak into the turbine. It might be worth while to try spiral grooves so cut as to tend to suck steam back from the lower pressure to the high.

As the steam passes from stage to stage its pressure falls and its volume increases. Hence we must increase

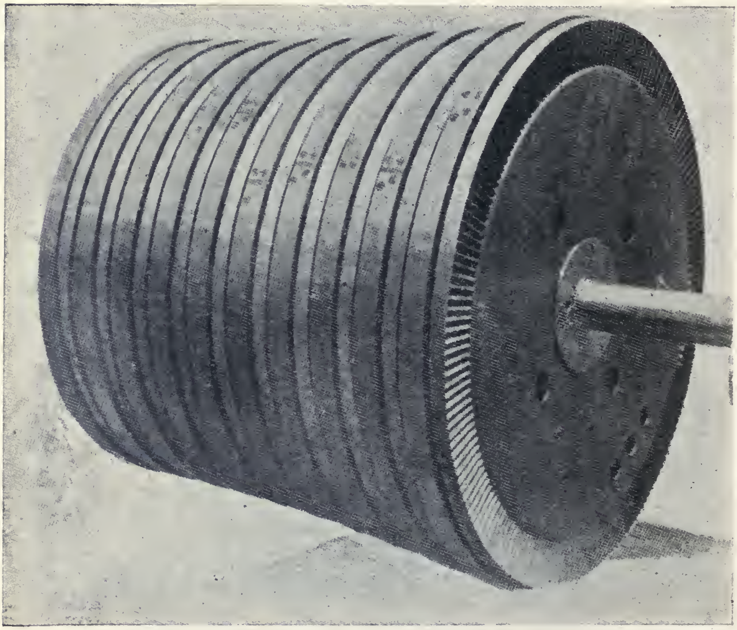

FIG. 16.—COMPLETE ROTOR, RATEAU STEAM TURBINE.

the nozzle areas. At first this is done by increasing the number of nozzles, and then by an increase in the wheel diameter and radial depth of the nozzles. The nozzles should be bunched together in a few symmetrically-placed circumferential arcs, so as to reduce the eddy losses at the boundaries of the streams of steam.

A throttle governor is used with (where desired) a by-pass which admits high-pressure steam lower down the turbine where the passage areas are greater, in case

of an overload. This is somewhat uneconomical of steam at overloads, but allows of a smaller turbine with its maximum economy at its normal full load. Fig. 14 gives a general sectional arrangement of this turbine, which is being manufactured by Messrs. Sautter, Harle, and Co., of Paris. Figs. 15 and 16 show views of a single wheel and the complete rotor with intermediate nozzle diaphragms, for a Rateau turbine. The shrouding over the blade tops is clearly shown, as is also the holes cut out of the wheels for balancing purposes. Fig. 17 is a

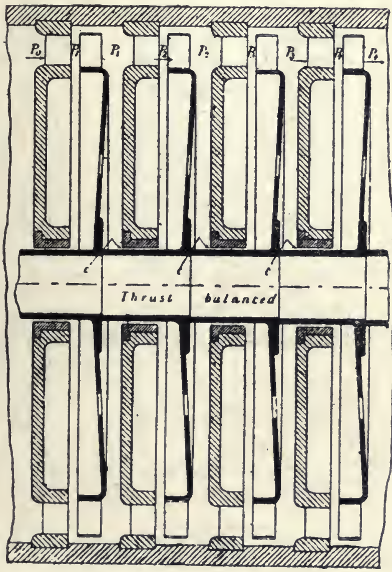

FIG. 17.—RATEAU MULTICELLULAR TURBINE.

partial sectional view showing the diaphragms and wheels rather more clearly than Fig. 14.

It will be noticed that Fig. 17 shows a number of holes in the rotating discs. The purpose of these holes is to insure that the steam pressure on both sides of a disc shall be the same, thus preventing end thrust due to an unbalanced pressure on the discs, and also insuring that practically the whole of the fall in steam pressure, with its accompanying generation of kinetic energy, shall take place in the nozzle diaphragms. These holes are

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usually made in the process of balancing the discs. They are seldom adopted when very high speeds, necessitating the maximum strength in the discs, are attained.

Fig. 18 shows one fo m of Rateau heat accumulator for use with a low-pressure turbine. The basins of water

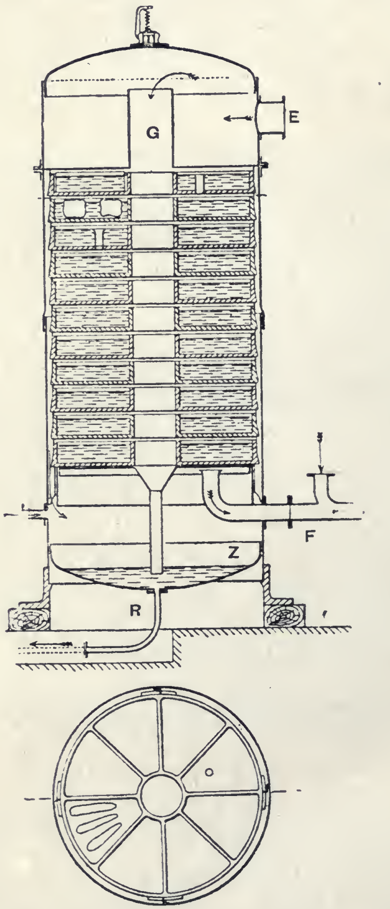


FIG. 18.—RATEAU HEAT ACCUMULATOR FOR EXHAUST STEAM TURBINE.

store heat extracted from the exhaust of high-pressure engines during light loads, and give out heat during overloads.

The steam from the high-pressure engines or turbines passes into the accumulator at *E* and partially condenses

on the trays. Steam for the low-pressure turbine is drawn off at F. If the load is a light one the pressure in the accumulator will rise and the temperature be also

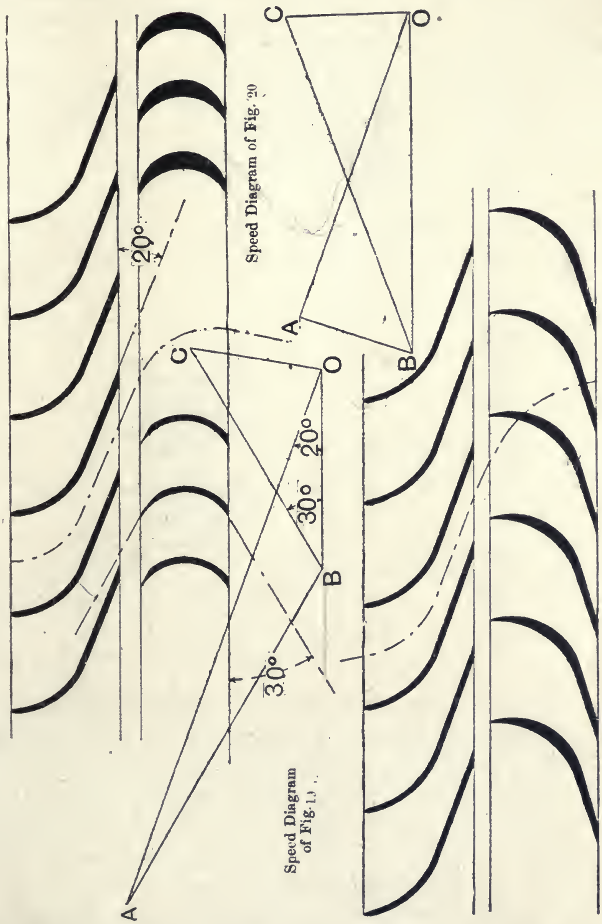


FIG. 19.—GUIDE VANES AND MOVING VANES OF AN IMPULSE TURBINE.

FIG. 20.—GUIDE VANES AND MOVING VANES OF A REACTION TURBINE.

raised, necessitating the condensation of some of the incoming steam. At overloads the pressure and temperature in the accumulator fall and consequently some of the water in the trays evaporates, thus securing a plentiful supply of steam to the turbine.

Fig. 19 illustrates the blade sections and velocity diagram for this turbine. The nozzle makes an angle of

20° to the wheel, the blades of which have angles of 30°. We shall see later that these angles are too small to give the most efficient results. Fig. 20 illustrates the blade shapes and steam velocities for a reaction turbine such as the Parsons. The differences between the two types of turbine are clearly brought out.

As in all impulse turbines, the axial clearances between the faces of the fixed and moving blades should be very small, especially so if the nozzles do not extend over the whole circumference of the diaphragm. If these clearances are large the moving steam becomes entangled with some of the stagnant steam filling the wheel chamber, and thus causes considerable eddy loss. This loss can be

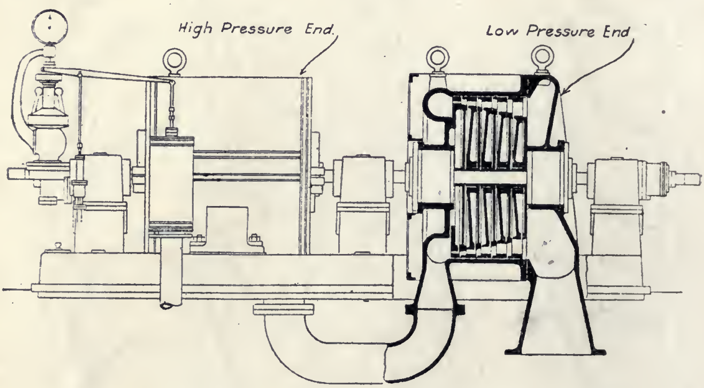


FIG. 21.—SECTIONAL ELEVATION OF ZOELLY STEAM TURBINE.

to some extent eliminated by mounting the moving blades directly on a drum of full diameter, instead of on individual wheels mounted on a comparatively slender shaft. If this course is adopted, we, however, very considerably increase the leakage between the fixed diaphragms and the rotor, on account of the great circumferential length of the labyrinth packing (Fig. 13), so that the cure will probably be worse than the disease. The radial clearances between the blade ends and the casing are not of much consequence, which is fortunate, seeing that the long shaft used is not very rigid, and is liable to whipping.

Zoelly Turbine.—The Zoelly turbine belongs to the same general class as the Rateau. It has, however, two distinguishing features. The drop in pressure at the nozzles

is as large as possible without having recourse to a diverging nozzle. This reduces the number of stages to about 10, and is conducive to a high efficiency. A departure from standard practice has been made in the case of

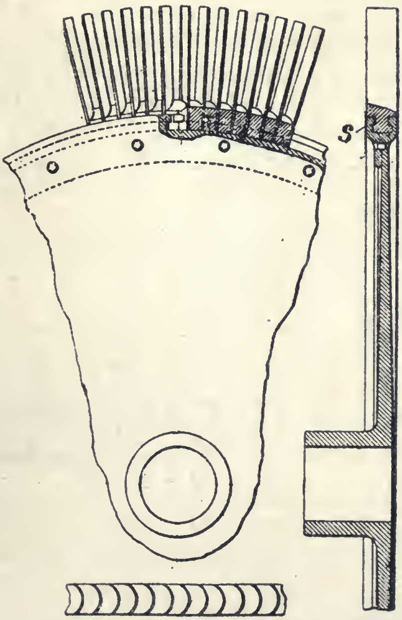


FIG. 22.—DISC OF ZOELLY TURBINE: LOW-PRESSURE END.

the moving blades. These are made with a greater radial depth at outlet than inlet, and hence permit of a small exit angle. This lack of symmetry in the moving blades gives rise to some end thrust, which is taken up by a thrust block. The wheels are of machined wrought steel.

The governor is of the throttle type with a steam controlling piston. The admission of steam to this piston is controlled by an auxiliary valve whose motion is regulated by a small rotary oil pump driven by the governor spindle. Fig. 21 gives a general view of the turbine.

According to Mr. Francis Hodgkinson,* in the later forms of the Zoelly turbine half the turbine (the high-

* "Some Theoretical and Practical Considerations in Steam Turbine Work." Proceedings Institution of Mechanical Engineers, June, 1904.

pressure section) is provided with tangential nozzles and buckets of the Pelton wheel type. The low pressure end has axial flow. The blades are very long—about half the wheel radius—and held in place on the wheels by two small projections on the blade. In this way the rim speed of the wheel proper is reduced. The blades are made very light and spaced further apart than is usual in turbine practice. No serious loss of efficiency is caused by this, it is claimed. These long blades must, however, greatly increase the friction of the wheels due to their rotation in a bath of steam. This is partly reduced by

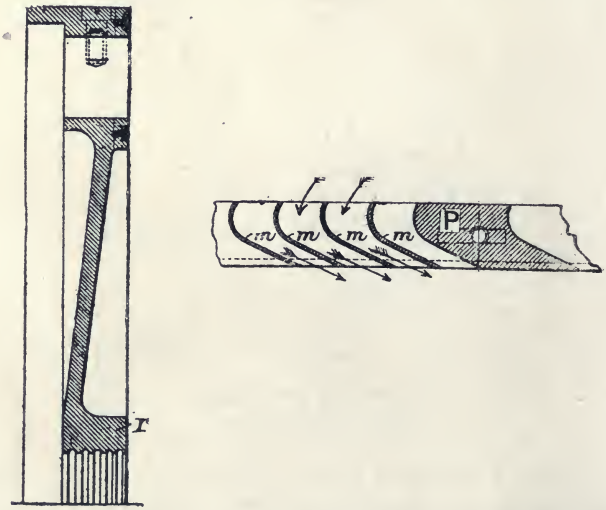


FIG. 23.—PARTITION AND NOZZLES OF ZOELLY TURBINE: LOW-PRESSURE END.

sheet steel housings at the sides of the blades. A shrouding is fitted.

Fig. 22 illustrates the parallel flow (low pressure) type of wheel, and shows how the blades are held in place by the clamping ring. It also shows the increase in the radial depth of the blades at outlet. Fig. 23 shows the construction of the diaphragms, whilst Fig. 24 shows a general sectional view of the turbine, and Fig. 25 illustrates the high-pressure blades, which are of the Pelton wheel type, the steam striking them tangentially. This turbine is now being manufactured in England by Mather and Platt, Ltd., of Manchester.

The turbine illustrated in Fig. 21 is the one now being built.

Lindmark Turbine.—This turbine may be of the reaction or impulse type. Its main feature is that the steam leaves the moving blades at a high velocity and is received

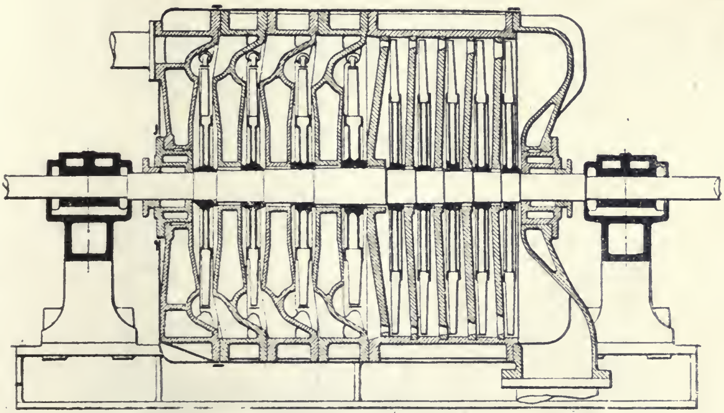


FIG. 24.—CROSS SECTION OF ZOELLY STEAM TURBINE.

in a diverging nozzle, which converts the kinetic energy (partially) into pressure energy ready for the next set of nozzles. The increase of pressure in the diverging nozzle is a source of considerable loss. The idea of converting

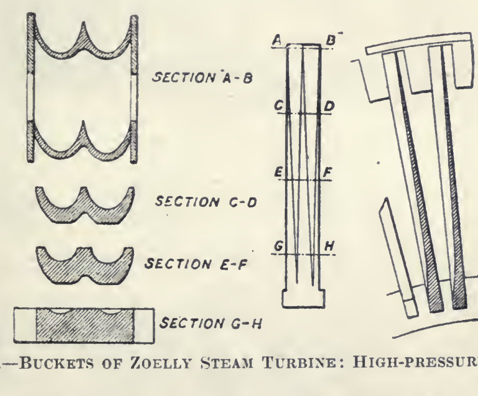


FIG. 25.—BUCKETS OF ZOELLY STEAM TURBINE: HIGH-PRESSURE END.

kinetic into pressure energy only to reconvert it into kinetic energy is a peculiar one, especially as the first conversion involves a considerable loss. Owing to the

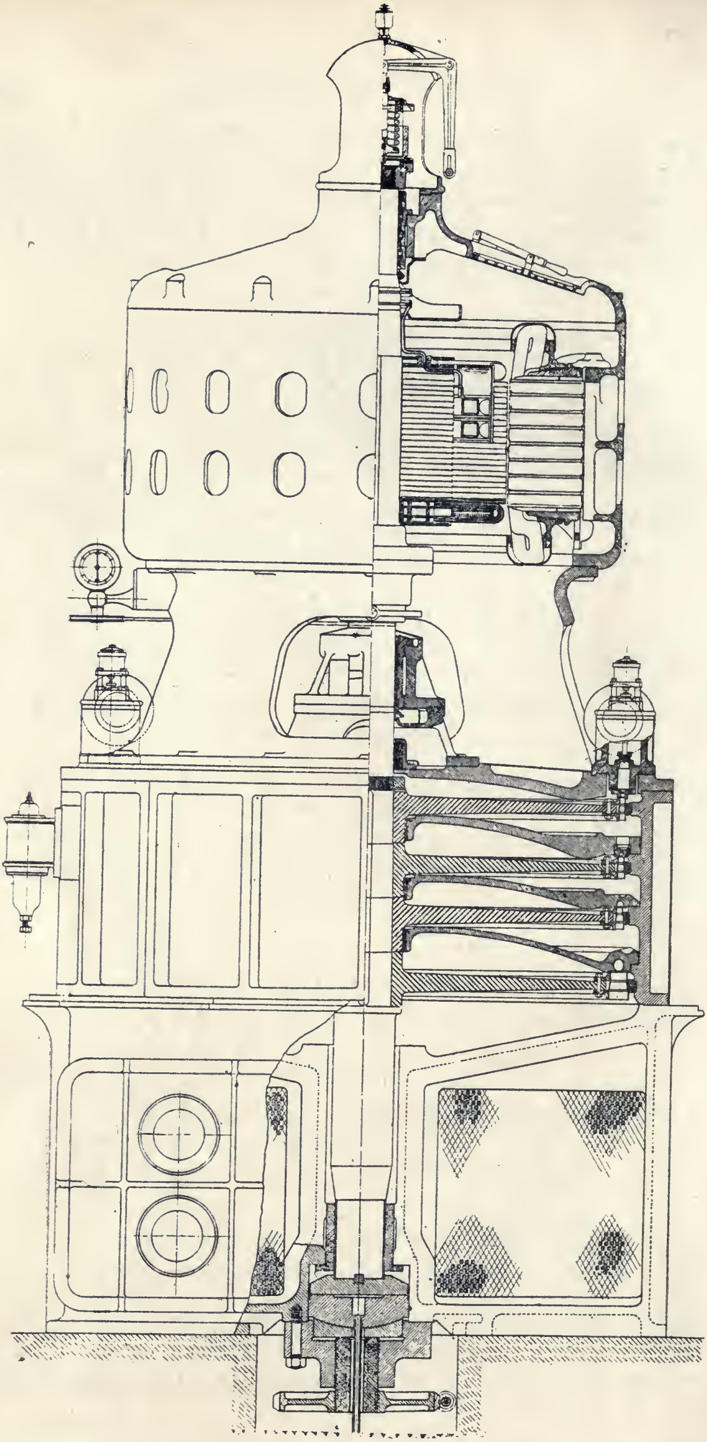


FIG. 26.—SECTION OF CURTIS FOUR-STAGE TURBO-GENERATOR WITH SUB-BASE CONDENSER.

high velocities used the work performed in each stage is large, thus requiring only a comparatively few stages.

Schultz Turbine.—This turbine is interesting mainly on account of the attempt made to get rid of end thrust. In the stationary turbine there are two cylinders (low and high pressure), the directions of the steam flow in them being in opposition. For marine work the high-pressure end is said to be of the impulse type. The thrust of the low-pressure end is said to balance that of the screw propeller.

Schultz also attempts to make the turbine capable of reversal. The direction of the flow of steam is reversed,

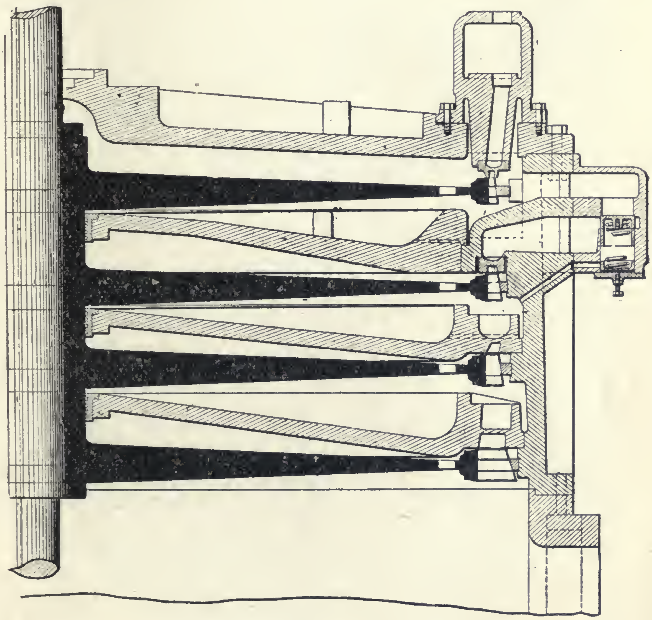


FIG. 27.—SECTION THROUGH CURTIS TURBINE.

and by means of diaphragms the passage areas are altered to suit the new conditions. This applies only to reaction turbines. We shall refer to this again when we come to deal with marine turbines.

Curtis Turbine.—As built in sizes above 400 kw., this turbine has a vertical shaft supported on a footstep

bearing. The generator is carried above the turbine. Fig. 26 gives a general idea of the turbine. There are four stages. In each stage there is a set of nozzles to generate kinetic energy, and two sets of moving blades

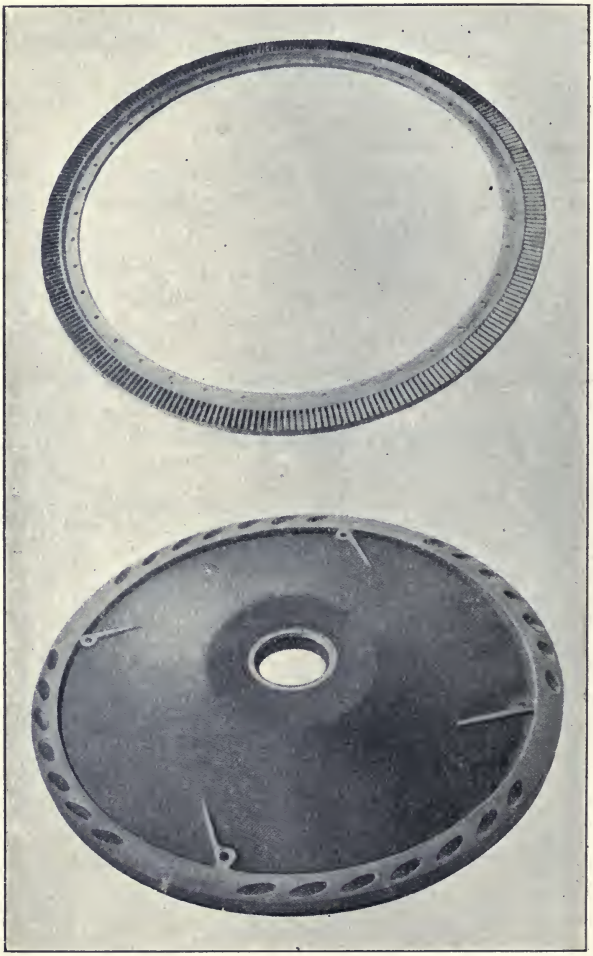


FIG. 28.—DIAPHRAGM AND REVOLVING BUCKET RING, CURTIS TURBINE.

with an intermediate set of stationary blades. Occasionally as many as four sets of moving blades per stage are employed, in which case there are usually only two stages. The greater the number of sets of moving blades in each stage the less the efficiency. The blades are

cut by a special shaping machine from a solid wheel or ring of steel. A slight projection is left on the centre of the blade end. A shrouding is fitted over the blade ends, and the aforesaid projection used to rivet the shrouding on.

Fig. 28 illustrates a diaphragm and a revolving bucket ring for a Curtis turbine. The bucket ring is, of course, attached to the rim of a wheel. For very large turbines the ring is sometimes in segments. In Fig. 29 is shown

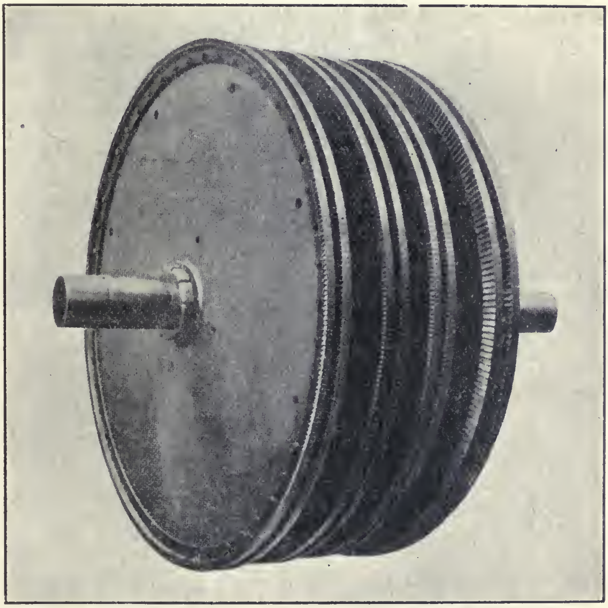


FIG. 29.—REVOLVING WHEELS OF FOUR-STAGE CURTIS TURBINE.

the complete turbine rotor consisting of four discs, each carrying two rings of blades. As in the Rateau turbine, the discs are pierced by a few holes for the purpose of insuring that the pressure on both sides the disc is substantially the same. In one 500 kw. Curtis turbine with two stages and three wheels—sets of moving blades—per stage there were 1,395 moving blades.

The radial clearances are of little consequence except where the shaft passes through the nozzle diaphragms. The axial clearances should be as small as possible.

They vary from 0·02in. in a 500 kw. machine to 0·08in. in a 5,000 kw. machine. The shrouding projects slightly

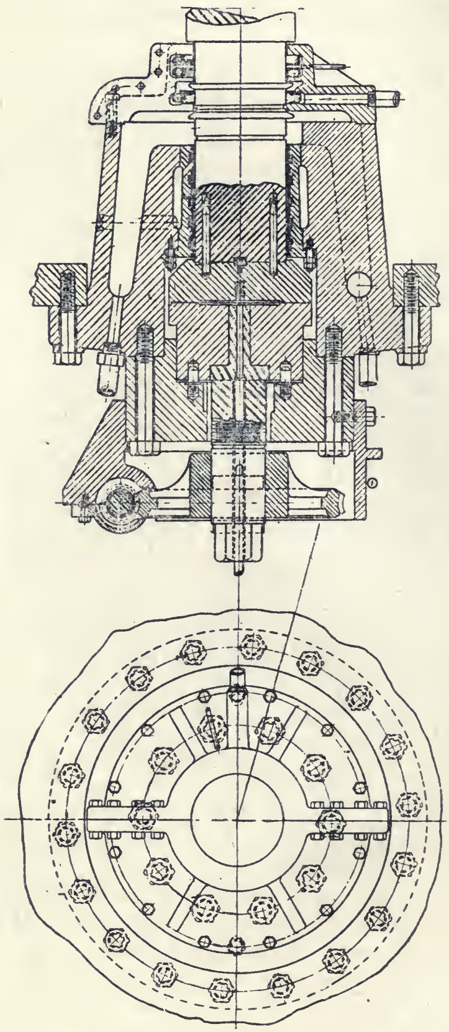


FIG. 30.—FOOTSTEP BEARING: CURTIS TURBINE.

beyond the edges of the blades, so that should the shaft sink a little the shrouding would come into contact with the turned face of the stationary diaphragm, and thus

protect the blades from damage. Some of the turbine rotors have a cast-iron brake ring attached. By applying brakes to this the turbine can be quickly stopped when the load is off. Otherwise it will run for four or five hours.

The footstep bearing (Fig. 30) has two bearing blocks, one attached to the end of the shaft, the other forming the bearing proper. Oil or water under pressure is brought up through the lower block and escapes between the upper and lower block, and then up between the shaft and a guide. Where oil is used a stuffing box and gland is

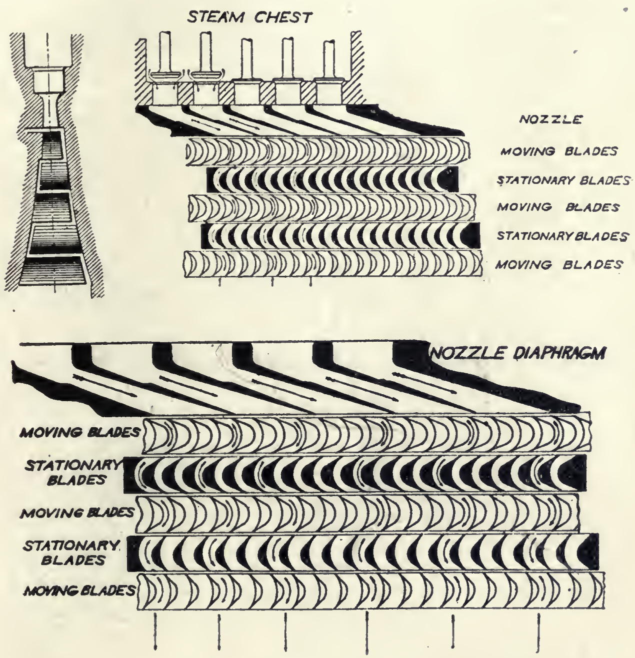


FIG. 31.—STEAM PATH IN CURTIS TURBINE.

required to keep the oil out of the exhaust chamber. With water this is not necessary. The water (or oil) pressure varies from about 175lbs. per square inch for a 500 kw. machine to 900lbs. per square inch for the 5,000 kw. turbine. This fluid pressure carries the weight of the turbine and generator rotors, which thus float on oil, and the bearing friction is consequently small.

Unlike a horizontal turbine, however, the bearing friction (so far as the footstep bearing is concerned) will increase, roughly, as the velocity, so that the friction which occurs when the rotor is just turned round is a very poor guide to the friction under normal running conditions.

In order to prevent a failure of the water supply to the footstep bearing duplicate pumps or accumulators are

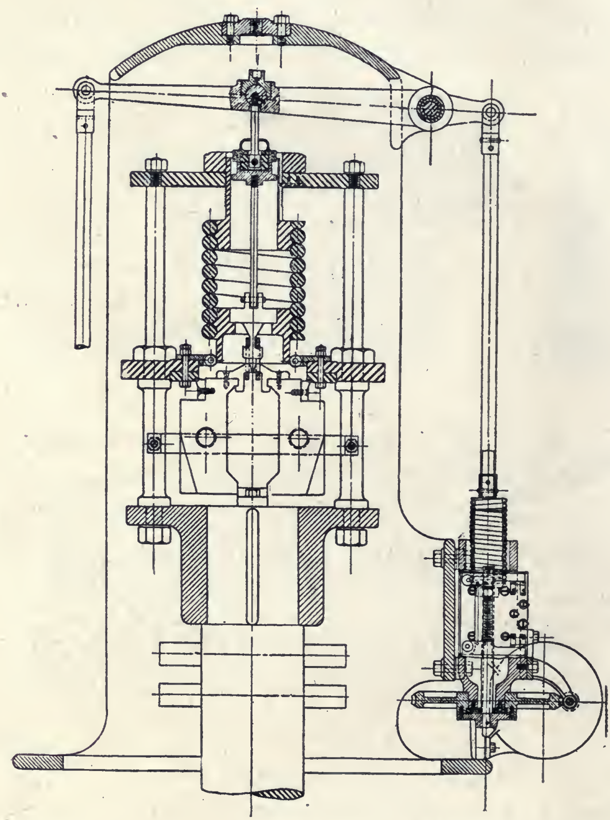


FIG. 32.—GOVERNOR FOR 500 KW. CURTIS TURBINE.

frequently used. The lower bearing block is adjustable vertically by means of a worm and wormwheel nut. This enables the axial clearances to be adjusted. Sight

holes in the turbine casing are provided to assist this adjustment of the clearances.

The governor is connected to a small electric controller. This controller supplies current to small electro-magnets,

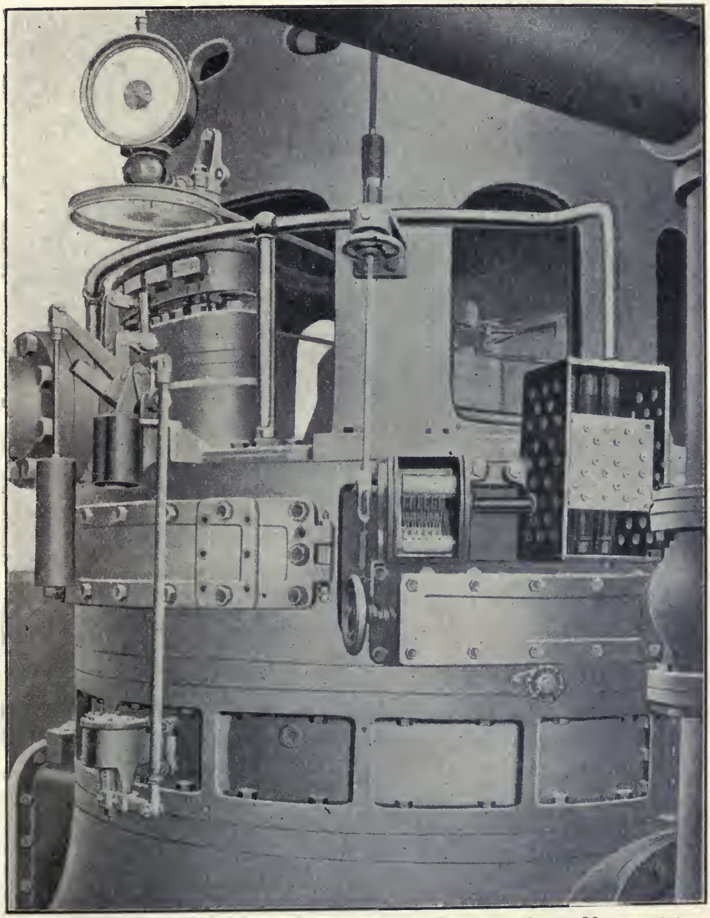


FIG. 33.—VIEW OF 500 KW. CURTIS TURBINE, SHOWING VALVE MECHANISM, CONTROLLER, AND RHEOSTAT.

which in turn open or close small steam valves. These valves admit steam to either side of a series of balanced admission valves, one for each nozzle in the first stage. In this way the number of nozzles which are open is made

to depend on the load. As, however, this only applies to the first stage, it is clear that the steam-pressure distribution at light loads in the other stages will be imperfect. An emergency governor is provided to prevent the turbine running away. No by-pass is used for overloads, the turbine being large enough to make this

TABLE II.

Size in Kw.	Speed, R.P.M.	Weight in Pounds.			
		Generator.	Turbine.	Rotor.	Whole Unit.
500	1,800	13,000	27,000	—	40,000
1,500	900	45,000	80,000	—	125,000
2,000	750	75,000	115,000	—	190,000
3,000	600	83,000	167,000	—	250,000
5,000	500	219,000	181,000	140,000	400,000
8,000	750	—	—	—	700,000

unnecessary. The condenser is frequently placed in a sub-base directly under the turbine. Table II. gives some data respecting Curtis turbines.

The 8,000 kw. turbine—which has to be capable of carrying an overload of 50 per cent. (a total load of 12,000 kw.)—is a new design for the Waterside station No. 2 of the New York Edison Company. It will be noticed in

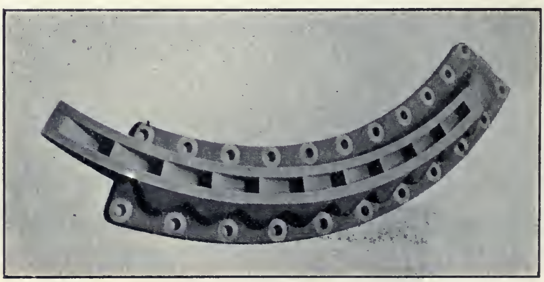


FIG. 34.—SET OF NOZZLES: CURTIS TURBINE.

particular that the rotational speed is considerably higher than that of the 5,000 kw. turbines which were previously

the largest turbines built by the General Electric Company. The diameter of the casing of this 8,000 kw. turbine at the base is a little over 15ft., and the height of the turbine 32ft. The step bearing is carried on a film of high-pressure water.

The usual peripheral velocity of the blades is about 420ft. per second. The 5,000 kw. turbines for Chicago had a peripheral velocity of only 320ft. per second on account of the difficulty of transporting larger diameters of wheels.

Fig. 31 gives a diagrammatic illustration of the steam path through the turbine, and shows also some of the first-stage nozzles closed. Fig. 32 shows the governor for a 500 kw. turbine. The two weights below the spring tend to fly out as the speed rises, and in doing so depress the lever passing through the governor casing at the top, which rotates the controller drum (Fig. 33). Fig. 33 also shows the emergency shut-off gear which breaks the vacuum as well as cuts off the steam supply. Fig. 34 shows a portion of a nozzle for a large turbine. The division walls in the nozzle are of sheet steel cast in position.

Figs. 35 and 36 show the sub-base condenser for one of the 5,000 kw. Chicago turbines (one of the earlier designs). There are four sets of tubes in series, each set consisting of about 560 tubes of 1in. diameter and 14ft. 3in. long. The objections to a sub-base condensers are that it is not quite so accessible as a separate condenser, and that it is liable to distort the casing of the turbine by altering the distribution of temperature. Fig. 37 gives the general over-all dimensions of a 5,000 kw. turbine.

Riedler-Stumpf.—There are several forms of this turbine. One form is identical with the Curtis in principle; another is similar to the Rateau in principle, but has much higher rim speeds. Still another is a simple De Laval in principle, but has wheels of large diameter.

By making the turbine wheels of large diameter comparatively low rotational speeds are obtained at the same time as a high peripheral speed. This makes for a comparatively simple machine. The wheels are made of a 10

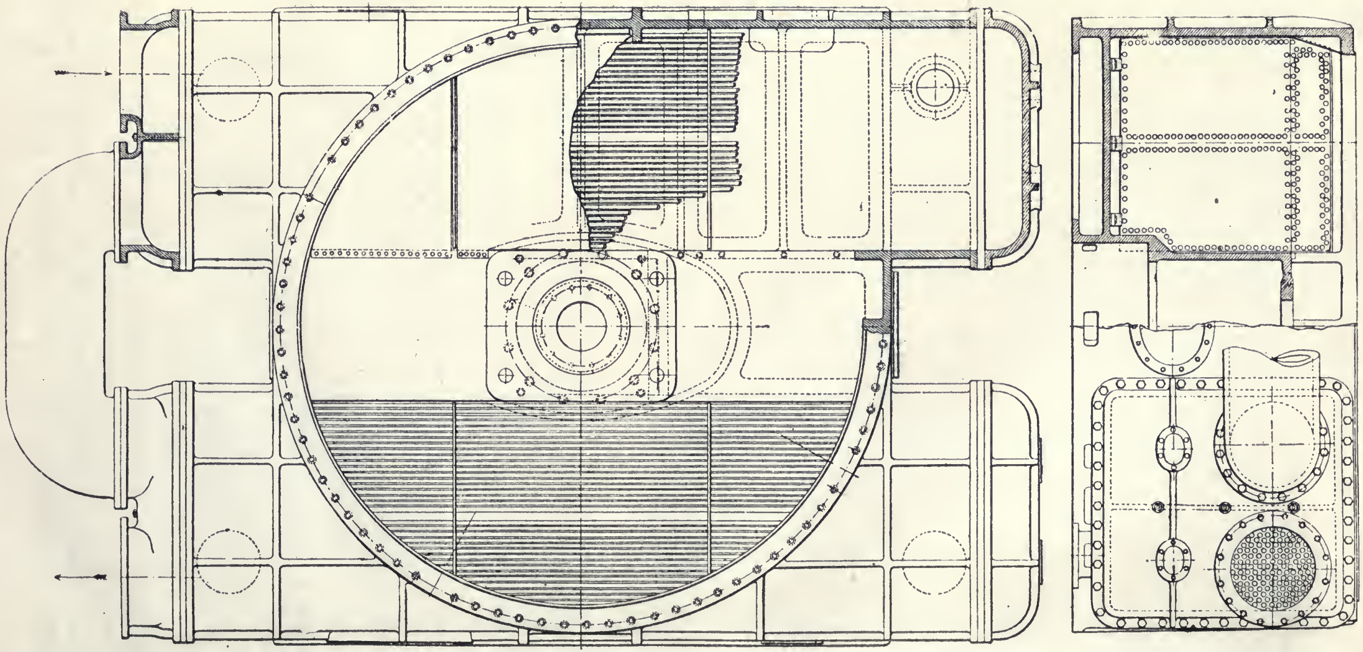


FIG. 35.—SUB-BASE CONDENSER 5,000 KW. CURTIS TURBINE.

per cent. nickel steel with a tensile strength of 135,000lbs. per square inch. In one turbine of 2,000 h.p. a rim speed of 1,031ft. per second was used, the maximum

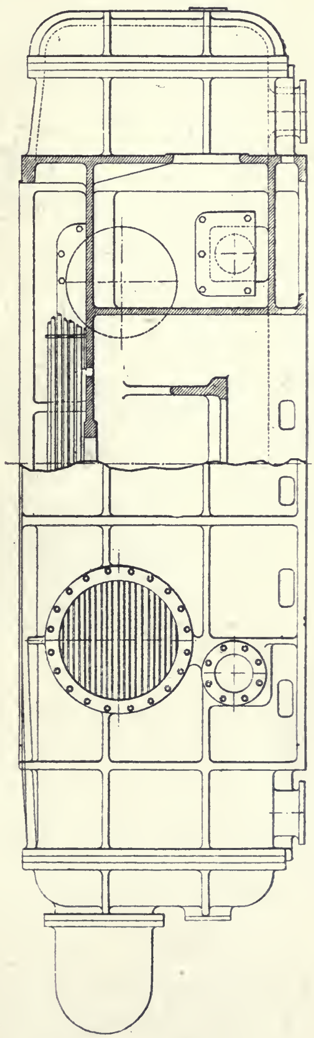


FIG. 36.—SUB-BASE CONDENSER 5,000 KW. CURTIS TURBINE.

stress in the wheel being 27,000lbs. per square inch. The wheel was 6·5ft. diam., and ran at 3,000 revs. per minute.

In this case there was only one wheel, the steam being directed on to the wheel twice.

The buckets of this turbine are illustrated in Fig. 38. They are of the Pelton wheel type, milled out of the solid rim. They overlap one another very like slates on the roof of a building (Fig. 39).

The following table (Table III.) gives the speeds of

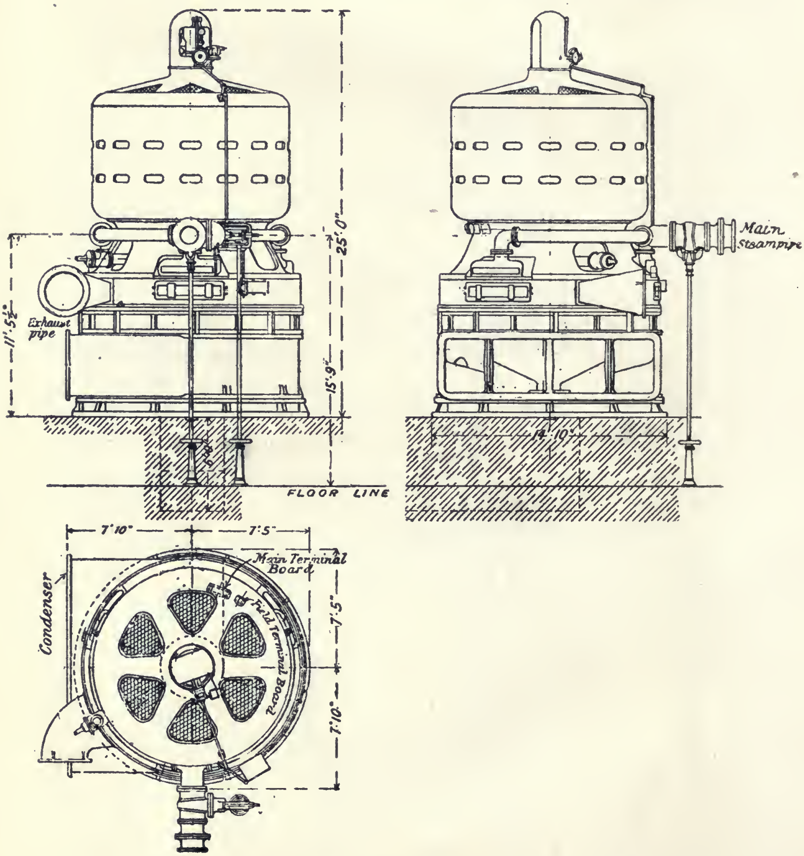


FIG. 37.—5,000 KW. CURTIS STEAM TURBINE, OVERALL DIMENSIONS.

these turbines for driving alternators and continuous-current machines. The speeds of the former are to a considerable extent restricted by the necessities of the alternator design.

TABLE III.

Kw. Size.		50	100	200	500	1,000	2,000	5,000
Alternating ...	speed in R.P.M.	—	3,000	3,000	3,000	3,000	1,500	750
Continuous current ..	speed in R.P.M.	3,500	3,000	2,300	1,500	—	—	—

The rotor buckets not played upon by steam are covered over by the casing being brought to within about 3 mm. so as to reduce the disc friction. The clearance between the rotor and the nozzles of a 2,000 i.h.p. turbine was 3 mm. The wheel weighed 1,875lbs. and its centre of gravity was brought—by balancing—to within 0·01 mm.

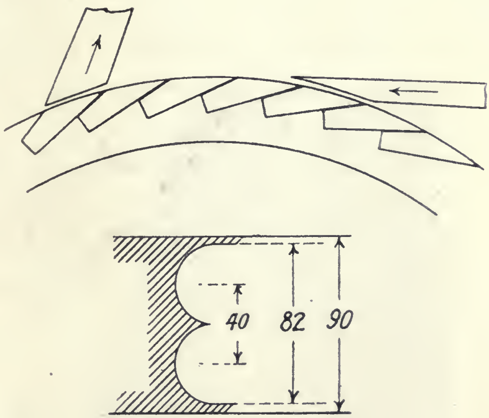


Fig. 38.—Buckets of Riedler-Stumpf Turbine.

of the axis of rotation. The wheel had 150 cut pockets of the size shown in Fig. 38. The nozzles are square ended, so as to give a better jet of steam, one that fits into the pockets (buckets) better than a round jet.

The most obvious objections to this type of turbine are the high speed and the excessive amount of steam friction in the guide channels conveying the steam from one part of the wheel to another. The turbine is being manufactured by the Allgemeine Elektricitäts-Gesellschaft of Berlin. One or two general views are given illustrating the general arrangement of the turbine

(Figs. 40, 41, and 42). The turbine is usually a horizontal one.

The formation of the guide passages is shown in Figs. 44, 45, and 46. In Fig. 44 steam enters at *A*, at one side of the buckets, passes around the curved buckets of the wheel, as indicated by the arrow at *B*, and when it is discharged from the wheel buckets it enters the guide passages, is deflected at *C*, and finally enters the wheel buckets again at the point *D*.

Figs. 45 and 46 show the method of conducting steam from the buckets of the first wheel to a second row of buckets of another wheel parallel and on the same axis with the first one. In Fig. 45 buckets similar to those of the Pelton water wheel are used, steam impinging at their centre and passing both to the right and left. It enters at *A*, is deflected by the buckets at *B B*, whence

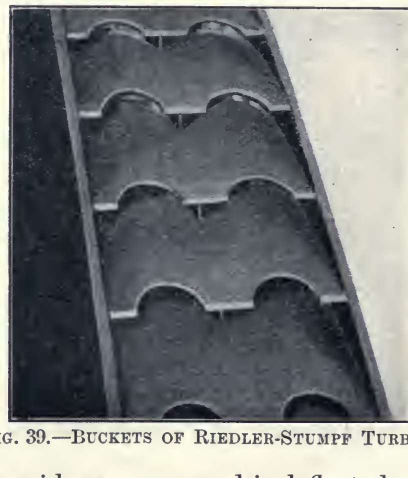


FIG. 39.—BUCKETS OF RIEDLER-STUMPF TURBINE.

it enters the guide passages and is deflected again at *C C*, and finally it impinges against the same row of buckets again in the direction of the arrow *D*. In Fig. 46 steam enters in the direction *A* at one side of buckets *B B*, where it is deflected and then passes around guide *C*, and finally impinges against another row of buckets in the direction *D* (see also Fig. 43).

Parsons Turbine.—A general view of the Westinghouse-Parsons turbine is given in Fig. 47. Steam enters

by the stop valve S and is admitted to the cylinder at A by means of the balanced valve V, which is under the control of the governor. The steam flows in an approxi-

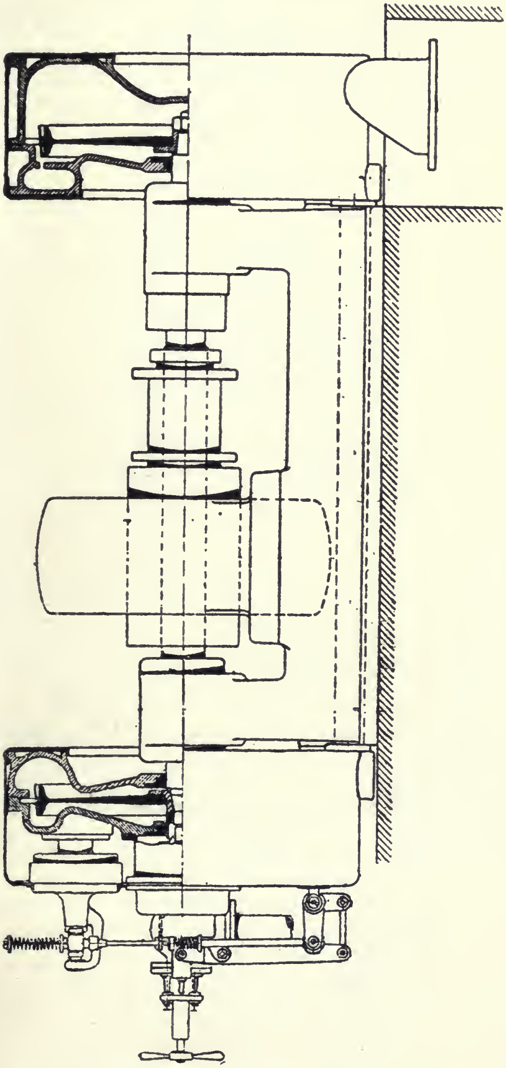


FIG. 40.—GENERAL ARRANGEMENT OF RIEDLER-STUMPF STEAM TURBINE.

mately axial direction from A to the exhaust B. Three different diameters are used. Otherwise the passage areas at the high-pressure end would be too small. The

steam velocity at the low-pressure end seldom exceeds 600ft. or falls below 160ft. per second at the high-pressure

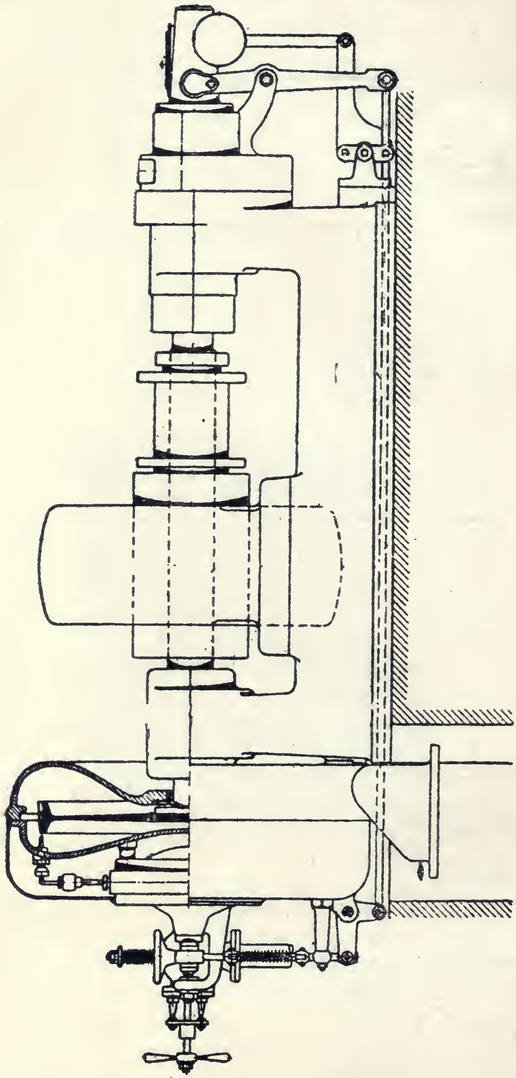


FIG. 41.—GENERAL ARRANGEMENT OF RIEDLER-STUMPF STEAM TURBINE.

end. The maximum peripheral speed of the drum varies from 300ft. up to (occasionally) 400ft. per second. The number of stages usually varies from 60 to 80. In a

1,500 kw. Westinghouse turbine running at 1,500 revs. per minute at the power-house of the Philadelphia Rapid Transit Company, the writer counted 72 stages. The minimum radial depth of the blades was about 0·75in.; the radial clearance varied from about 0·06in. to 0·1in. for the blades, and about 0·15in. for the rings on the balance pistons. These clearances would be somewhat

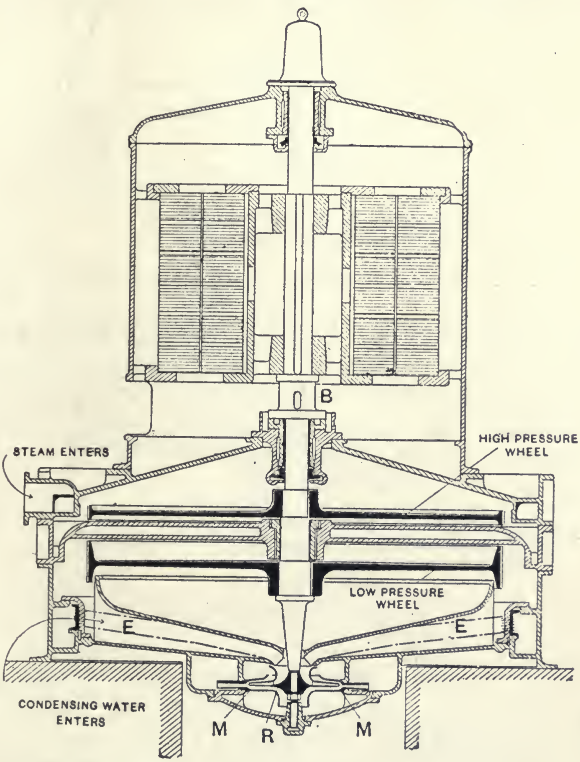


FIG. 42.—TWO-DISC RIEDLER STUMPF MACHINE WITH VERTICAL AXIS.

less when running. The axial clearances in the Westinghouse turbine are large, increasing from about 0·1in. at the high-pressure end to as much as ½in. at the low-pressure end.

The three pistons *P* shown in the figure rotate with the shaft. They are intended to balance the end thrust of the turbine. The pressures on their front ends correspond to the pressures at the beginning of the three

sections of blades. Their circumferences have a number of circular grooves cut into the cylindrical face. Dovetailed into the casing are corresponding brass projecting rings which protrude into the piston groove. The casing

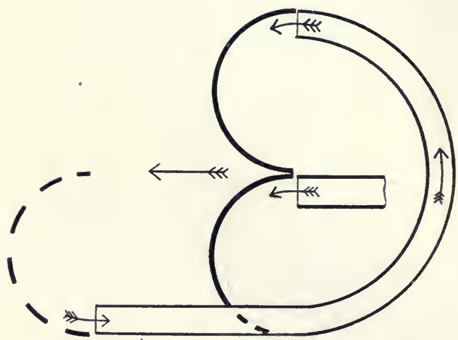


FIG. 43.—STEAM DISTRIBUTION IN RIEDLER-STUMPF TURBINE.

rings are made of brass strip about $\frac{1}{8}$in. thick and $\frac{1}{2}$in. wide. The strip has two small projections in the direction of its length. One of these projections serves

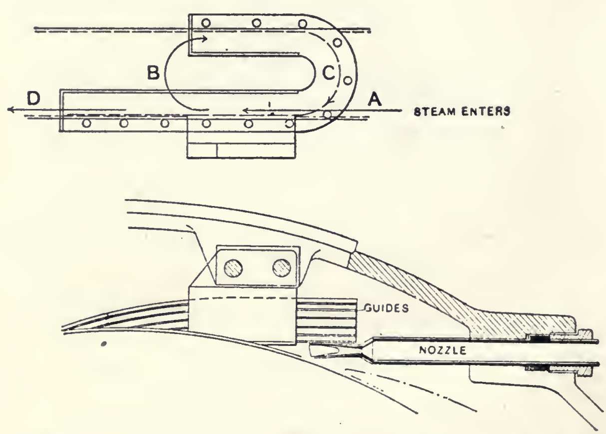


FIG. 44.—METHOD OF APPLYING STEAM TWICE TO THE SAME SET OF BUCKETS.

to hold the strip in the casing. The other very nearly touches one of the radial surfaces of a piston groove. With such an arrangement very little leakage of steam over the circumference of the piston can take place, and

there is no metallic contact between rubbing surfaces. The piston grooves are about 0·25in. wide.

For fixing the blades in position, the following is a common method. Shallow grooves, from 0·25in. to 0·4in. deep and the correct width to take the blades, are turned in the rotor or casing. The grooves are slightly dovetailed. The blades, which may be of brass, delta metal, or copper, are frequently nicked at the groove end. The

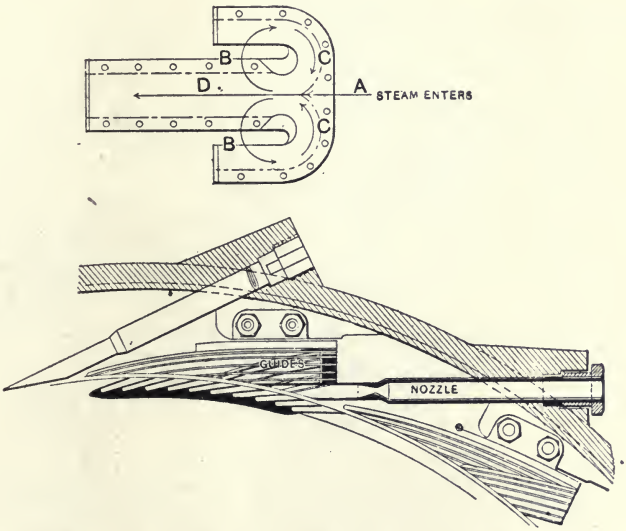


FIG. 45.—METHOD OF APPLYING STEAM TWICE TO THE SAME SET OF BUCKETS WHEN BUCKETS ARE OF THE PELTON TYPE.

blades are placed radially in the grooves, alternating with filler or caulking pieces of soft-drawn brass. The latter are caulked and hold the blades firmly in position. For the longer blades support is given to the outer ends by passing through and soldering to them a wire.

For adjusting the axial position of the rotor and taking up any small unbalanced end thrust a small thrust block T is provided. This is in halves adjustable separately in opposite directions, so as to limit the axial motion of the turbine rotor.

The governor is usually of the centrifugal ball type, and is illustrated in Fig. 48. It admits steam to the

cylinder in blasts, the duration of which is increased with the load. To enable an overload to be taken the governor will open a second admission valve V_2 which admits steam at a point in the casing where the passages are larger than at the (normal) high-pressure end. The ball levers of the governor are mounted on knife edges to increase the sensitiveness. The rod B is driven by worm gearing—which also drives the oil pump—from

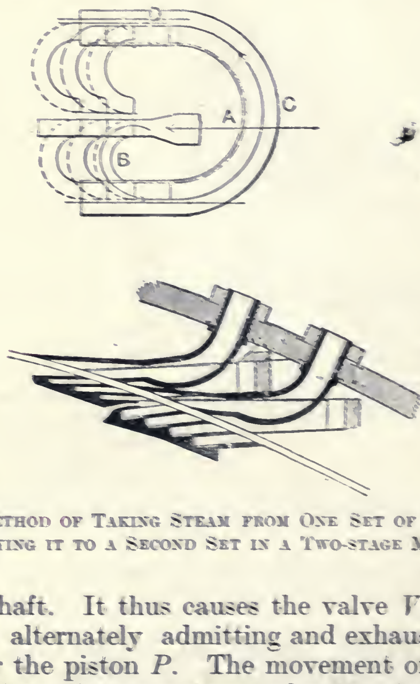


FIG. 46.—METHOD OF TAKING STEAM FROM ONE SET OF BUCKETS AND CONVEYING IT TO A SECOND SET IN A TWO-STAGE MACHINE.

the main shaft. It thus causes the valve V_1 to reciprocate, thus alternately admitting and exhausting steam, from under the piston P. The movement of this piston opens and closes the main steam valve V. As the speed of the turbine rises the lever A is raised, carrying with it the small piston valve, and brings about an earlier exhaust from the auxiliary steam cylinder, thus reducing the length of the steam blast admitted to the turbine.

A small plunger pump supplies oil under pressure to the bearings. The oil pressure at the pump varies from 3lbs. to 15lbs. per square inch, and is intended to overcome the frictional resistance of the distributing pipes and passages.

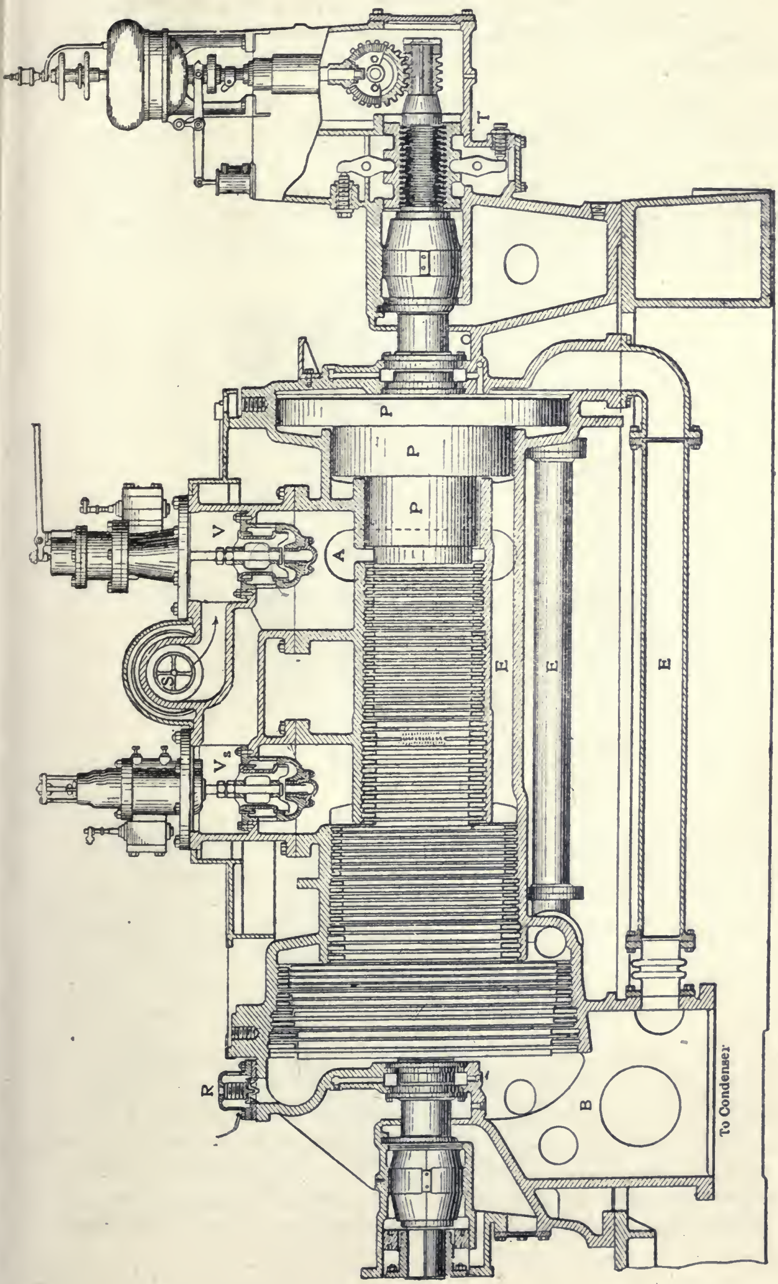


FIG. 47.—SECTION OF WESTINGHOUSE-PARSONS TURBINE.

Fig. 49 illustrates a 5,000 kw. Westinghouse-Parsons turbo-generator capable of carrying an overload of 50 per cent.

British Westinghouse.—This turbine embodies several departures from established practice. It is double-ended, the steam entering at the middle point of the cylinder and flowing in two streams, one toward each end. This produces a balance on the rotor, there being no appreciable end thrust. The turbine is a combination of the impulse

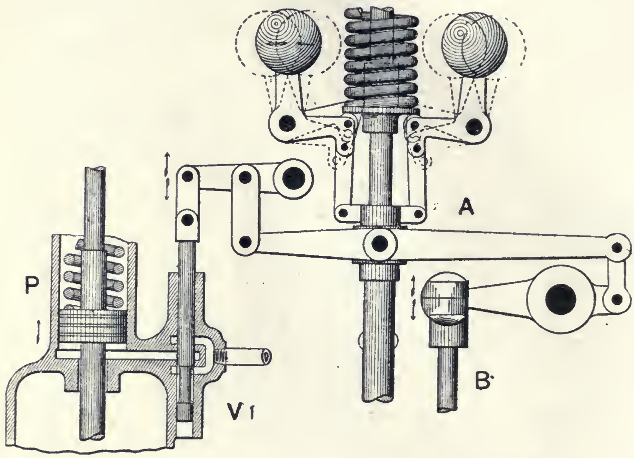


FIG. 48.—WESTINGHOUSE-PARSONS TURBINE GOVERNOR MECHANISM. (See p. 44.)

and reaction types. The first fall in pressure, down to, say, 60lbs. absolute, is accomplished in the impulse section; the remainder of the expansion in the reaction section. The blade heights are, of course, only half those of a single-flow turbine of the same capacity, which is one reason for the adoption of the impulse high-pressure section where short blades are not of so much consequence.

The drum is of approximately uniform diameter, so that the peripheral speed is the same for all the blades. It will be remembered that in an ordinary Parsons turbine the peripheral speed of the blades at the high-pressure end is much less than that of the low-pressure blades. Consequently in the Parsons turbine the high-pressure blades do not absorb as much work per row as do the

low-pressure blades; whereas in the British Westinghouse turbine the work absorbed in each row is the same and equal to the maximum, thus making the number of

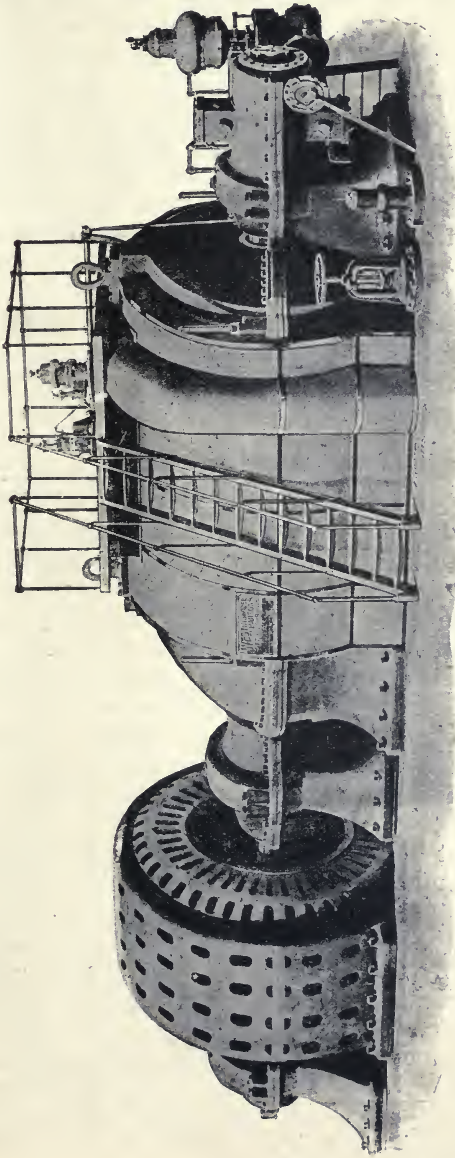


Fig. 49.—Westinghouse-Parsons 5,000 kw. Turbo Generator.

rows of blades or stages small. This gain is, however, largely counterbalanced by the fact that in this turbine there are virtually two complete turbines. The total number of stages or rows of moving blades is not usually very much less than in an ordinary Parsons turbine, being, in fact, usually between 60 and 70. The high-pressure impulse section is of the Curtis type, and contains (for each half of the turbine) a set of nozzles receiving steam from the annular steam chest (see Fig. 50), and discharging into two sets of moving blades mounted on the rotor, with an intermediate set of fixed guides mounted on the casing. Then follow the reaction blades. The impulse blades have a shroud over their ends, the reaction blades being open ended.

Owing to the absence of the dummy or rotating balance pistons the length of the rotor between bearings is somewhat shorter than in the ordinary Parsons turbine. It could be made still shorter if the turbine were made single instead of double flow. In this case balance pistons would have to be provided, but there would be the distinct advantages of having the blade lengths doubled, the number of rows of blades halved, and the rotor length reduced. The exceptionally small blade heights at the high-pressure end of the turbine, due partly to the fact of the turbine being double flow, but mainly to the high peripheral and steam velocities adopted for these blades, necessitate the radial clearances being a large percentage of the blade heights, and hence cause a very large clearance leakage, which considerably reduces the efficiency of the turbine and makes the adoption of this type of turbine well-nigh impossible for powers of less than 500 kw. or even higher.

The thrust block for adjusting the relative positions of the rotor and casing must be outside one bearing, and hence owing to the unequal expansion of the rotor and casing there must be a considerable axial clearance between the fixed and moving impulse blades, which in the absence of full peripheral admission from the nozzles is not desirable.

The general constructive features of this turbine * are illustrated in Fig. 50. The shaft is of high-carbon steel

* "Mechanical Construction of Steam Turbines and Turbo-generators," by W. J. A., London. Proceedings of Institution of Electrical Engineers, 1905, part 173; vol. 35.

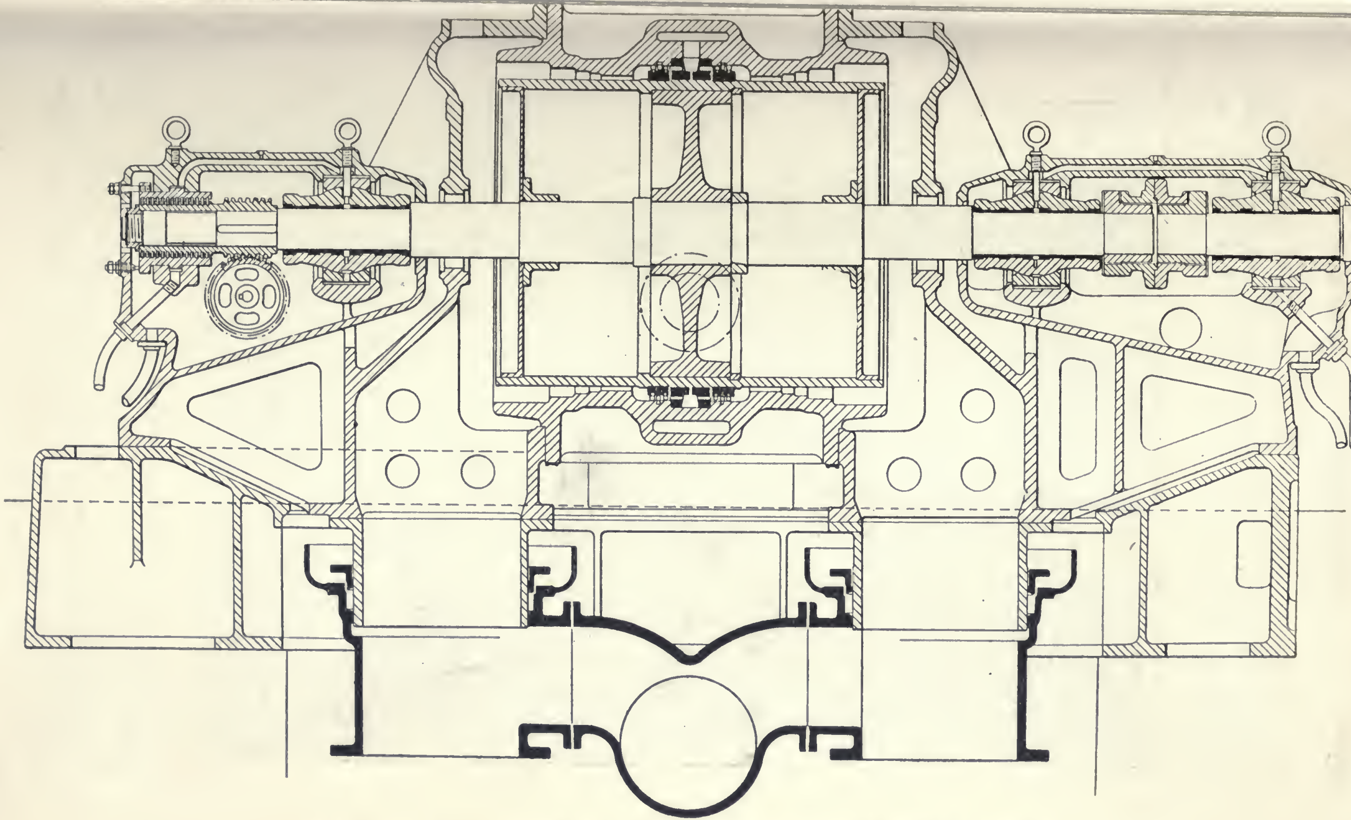


Fig. 50.—Longitudinal Section of British Westinghouse Steam Turbine.

and carries a forged-steel disc for carrying the weldless rolled steel drum. This drum is further supported at its ends by steel end plates.† The impulse blades are in steel rings shrunk on to the drum.

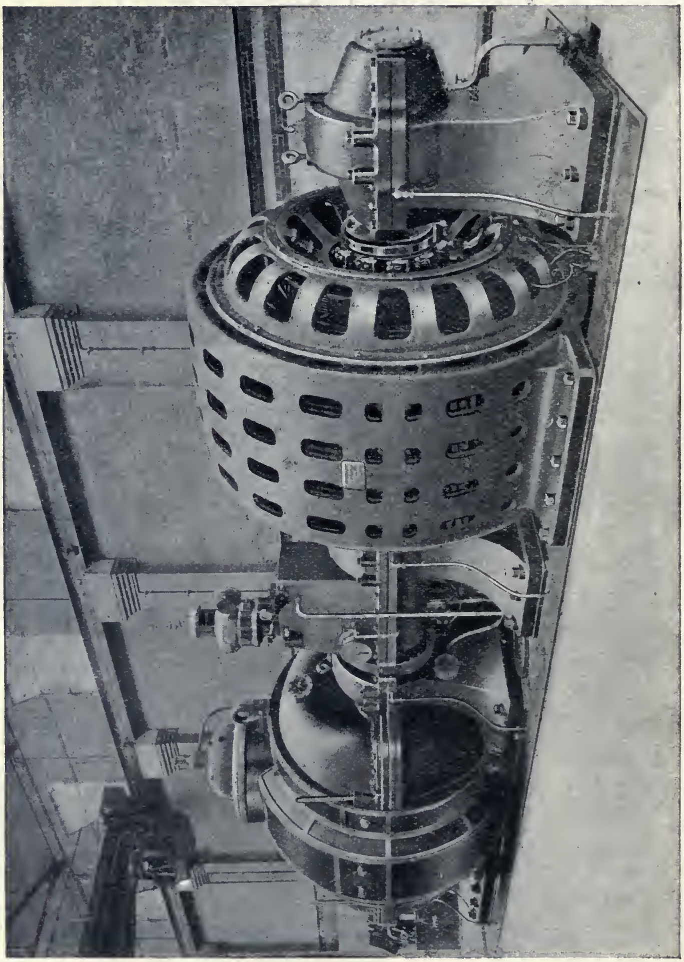


FIG. 51.—3,500 KW. BRITISH WESTINGHOUSE TURBO-ALTERNATOR.

The British Westinghouse turbines, supplied to the London Underground Railway Company, are of 5,500 kw. capacity each, with a speed of 1,000 revolutions

† These thin steel end plates have been replaced by much more solid end plates.

per minute. The turbine casing is bolted to the bed plate at the generator end, the other end sliding on the machined surface of the bed, so as to allow for unequal expansion of the casing and bed. The bearings—which each weigh about eight tons—are water cooled. The generator is 3-phase, 11,000 volts at $33\frac{1}{3}$ cycles per second. Its weight is about 90 tons, of which the armature accounts for 53 tons. The surface condenser for each turbine has a cooling surface of 15,000 square feet, or nearly 3,000 square feet per kw. capacity. The circulating water is taken from the river Thames, and is at a comparatively low temperature, so that the condenser is not so large as would usually be considered necessary where the temperature of the water was higher.

The British Westinghouse turbines for the Clyde Valley Power Station are of 2,000 kw. capacity each, capable of taking an overload of 50 per cent. The revolutions per minute are 1,500, and the blade speed—or peripheral velocity of the drum—is stated to be five miles a minute, or about 440 feet per second, which seems much too high. The blades at the high-pressure end are of drop-forged steel. In all, there are about 20,000 blades in each turbine.

Reuter Turbine.—The accompanying illustrations show a design of 2-stage turbine, the invention of Theodore Reuter, Winterthur, Switzerland. Fig. 52 is a vertical section; Fig. 53 is a front elevation, partly in cross section, omitting for the sake of clearness the outer ring of pressure pipes and cylinders; and Fig. 54 is a longitudinal section drawn to a larger scale through the valve-operating device.

The turbine has two chambers A and A^1, in each of which are placed alternately a running apparatus and a guide apparatus. The driving medium is supplied to the first stage A by means of a ring of nozzles B, to which the driving medium is conveyed by means of an annular inlet passage C. The several nozzles are closed towards this annular passage by means of small valves D. The stage A^1, arranged behind the stage A, is provided with a ring of nozzles B^1 through which the driving medium flows to the second stage A^1 after it has passed the first stage A. In front of these nozzles B^1 there are arranged closing parts D.

The control of the valves or similar closing parts is effected by means of a pressure liquid. The valves D D^1

are connected with a piston E or E^1 (Fig. 52), which moves in a small cylinder F or F^1, and against which

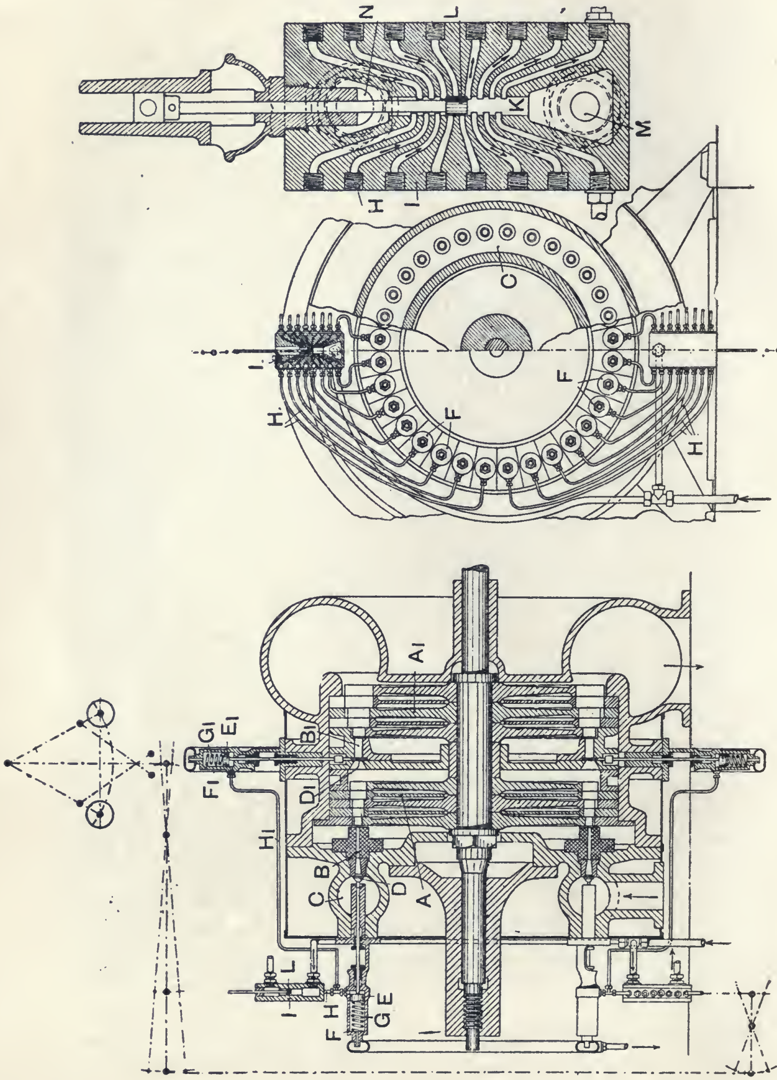


FIG. 54.

FIG. 53.

FIG. 52.

REUTER'S TWO-STAGE STEAM TURBINE.

there presses a spring G or G^1 in such a manner as to close the corresponding valves when there is no pressure

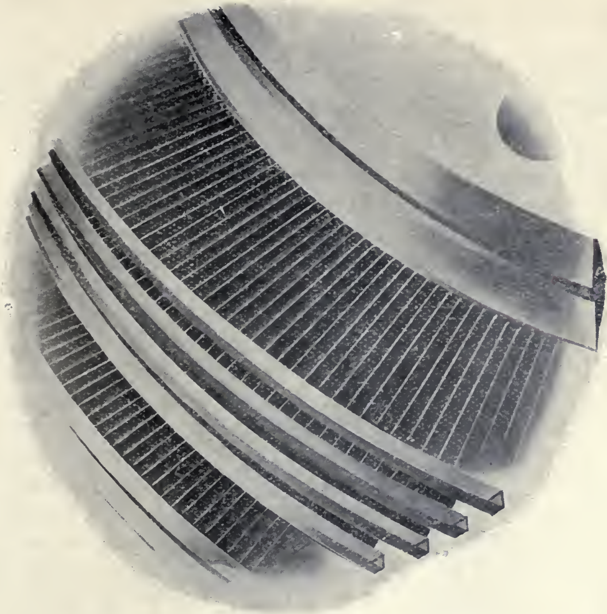
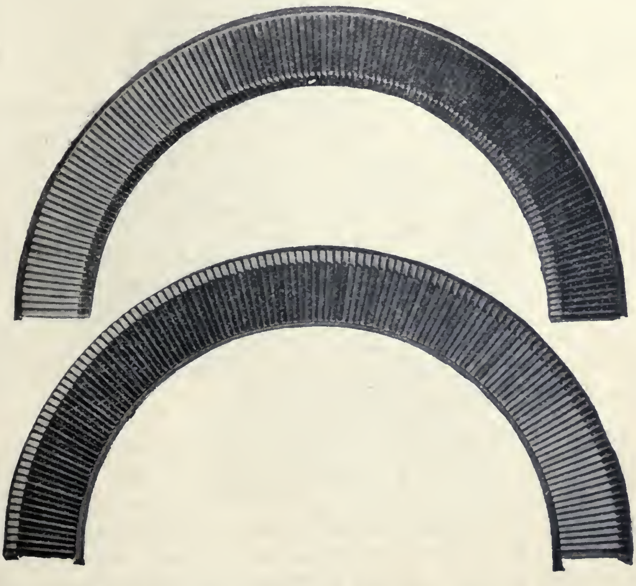


FIG. 55.—BLADING OF WILLANS & ROBINSON TURBINE.
ROTOR BLADES.



CASING BLADES.
FIG. 56.

liquid admitted by a pipe H or H^1 behind the piston, which would open the valves D or D^1 in opposition to the spring G or G^1.

The pipes H lead, as shown more clearly in Fig. 53, from the cylinders F to the controlling devices I, and also the pipes H^1 from the cylinders F^1, where they open into a cylindrical passage K in which moves a piston L. The latter is extended outwardly in the form of a rod which, as shown diagrammatically by dotted lines in Fig. 52, is under the influence of the governor. The cylindrical passage K of the controlling device I communicates at one end with the supply M, Fig. 54, and at the other end with the outlet N of the pressure liquid. As shown in Fig. 54, according to the position of the piston slide L there are more or less pipes H in communication with the pressure supply M, thus leading the pressure liquid to their respective cylinders F, whereby the respective closing parts D are open. As the pipes H^1 are branched off from the pipes H, one and the same controlling devices I will also open, in addition to those of the first stage, a corresponding number of inlets to the second stage. The passages H opening between the piston L and the pressure outlet N, draw away the pressure liquid from the corresponding cylinders F and F^1, so that the respective valves are closed by the springs G and G^1.

It will be seen that this turbine consists essentially of a horizontal 2-stage Curtis turbine with some differences in the constructive features, more particularly in respect of the diaphragms containing the guide blades, and in the method of governing. It is this last which is the most interesting feature of the turbine, particularly the arrangement whereby the number of nozzles in *both* stages is varied simultaneously in order to maintain a correct steam distribution at all loads. In the Curtis turbine it will be remembered only the number of nozzles in the first stage is varied.

Willans & Robinson. — Messrs. Willans & Robinson, Ltd., of Rugby, have recently come to the front as manufacturers of a turbine of the Parsons type. The chief feature of their turbine is the blading system employed. The blades themselves are of brass, or of a copper-nickel alloy at the high-pressure end when high superheat is used, cut from strips drawn to the correct

section. A light channel section of the same axial width as the blades, and about three or four millimetres

FIG. 57.—ROTOR OF 3,000 KW. WILLANS & ROBINSON TURBINE.

radial depth, is riveted over the ends of the blades, the blade material itself being used for riveting. The sides

of the channel are of course turned away from the blade. The idea of this channel is twofold. In the first place it is hoped that it may prevent leakage to a considerable extent by reason of its shape, and also because it enables the radial clearances to be considerably reduced. In a large turbine of this type the clearances varied between 0·025in. and 0·035in., being markedly less than in Parsons' turbines with the ordinary type of blading. The second advantage claimed for this type of construction—and the one which makes the first advantage possible—is that if the rotor and the casing should come into contact, then, save in very serious cases, no serious damage will be done, a little metal being worn off the channels and the turbine brought to a standstill.

The fixed ends of the blades are all fitted into a brass foundation ring (actually two half rings per set of blades) as is shown in Fig. 56. Each complete set of blades is in two semi-circular segments. These segments are laid in circular grooves (averaging about ½in. deep) and caulked tightly in position by means of a brass caulking strip which expands the foundation ring thoroughly into the slightly tapered groove. The whole construction is very solid and, as the channel and foundation ring are punched and cut to receive the blades by special machinery, the correct angular and relative positions of the blades are assured.

The shaft consists of a wrought-steel drum in one piece, with two end plates and short bearing shafts shrunk into the ends. The cylinder is, of course, split horizontally, but it is also in three or more (approximately) cylindrical sections, which reduces the size and weight of the individual pieces, a convenience in manufacturing and erecting, although not without its drawbacks.

Practically all the auxiliary gear is attached to or forms part of the support for the lower half of the turbine casing, making the dismantling of the turbine for inspection a comparatively simple matter, which is assisted by hinging the top half of the casing to the lower.

The governor is not a blast governor of the ordinary Parsons type, but is a simple throttle governor with a by-pass valve for overloads. An emergency governor is also fitted to shut down the turbine in case of a runaway. The low-pressure balance piston is fitted at the low-

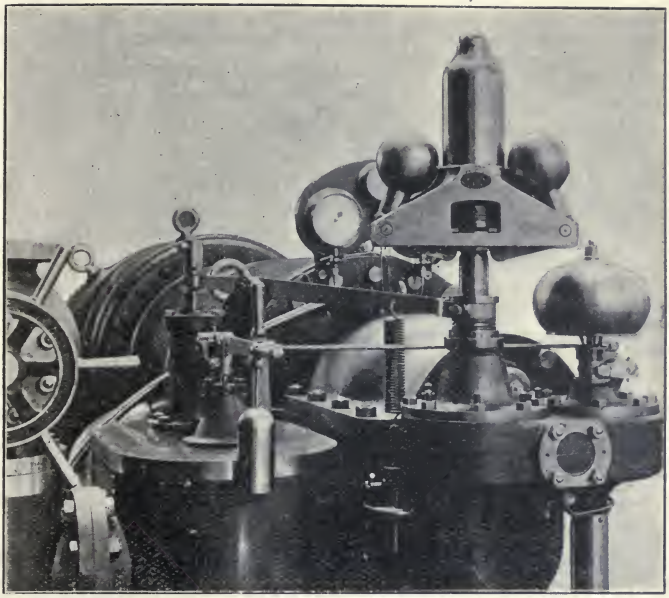


FIG. 58.

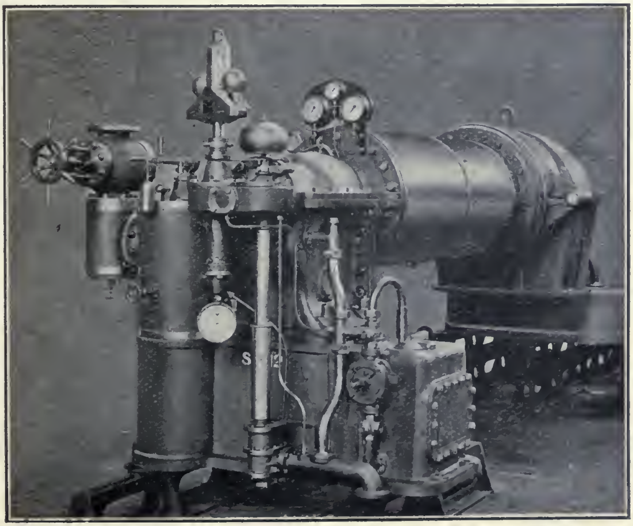


FIG. 59.

GOVERNOR GEAR OF WILLANS & ROBINSON TURBINE.

pressure end of the turbine. Fig. 57 shows the rotor of a 3,000 kw. turbine. The drum on which the blades are mounted has three sections of different diameter, the illustration showing clearly the increase in blade height as the exhaust end is approached. Figs. 58 and 59 illustrate the governor gear of a turbine supplied for the Bristol corporation. The large governor with the fly-balls is the Hartnel throttle governor. The smaller governor, fitted with a casing to prevent it being tampered with, is the emergency governor, whose function is to prevent the turbine speed exceeding a certain prearranged limit.

Fig. 60 shows a view of the Bristol turbine opened up for inspection. The upper half of the casing is hinged to the lower half so as to facilitate the opening up. It will be seen that the governor gear, being attached to the lower half of the casing, does not require removal.

The figure shows clearly the steam pipe leading from the main throttle-valve chest to the high-pressure end of the blading, and also the by-pass pipe leading from the lower valve chest to a point some little way along the turbine blading. The valve controlling the flow of steam along this latter pipe is on the same spindle as the main throttle valve and under the control of the same governor, but does not begin to open until the main throttle valve is admitting its maximum quantity of steam. The base tank at the high-pressure end of the turbine is used as an oil cooling reservoir.

The feature of this turbine which has given rise to the greatest amount of both praise and blame is the system of blading already described, and the very small radial clearances used. These small clearances were not adopted without first giving the matter careful consideration. Messrs. Willans & Robinson have performed a series of careful experiments on large turbines under running conditions with a view to determining the amount and nature of the distortion of turbine casings, and thus estimating approximately the minimum radial clearances which are necessary. The reduction of these radial clearances is one of the most important problems before the turbine designer, although more especially so in small turbines, as the length of blades in large-power turbines reduces the radial clearance considered as a percentage of the blade

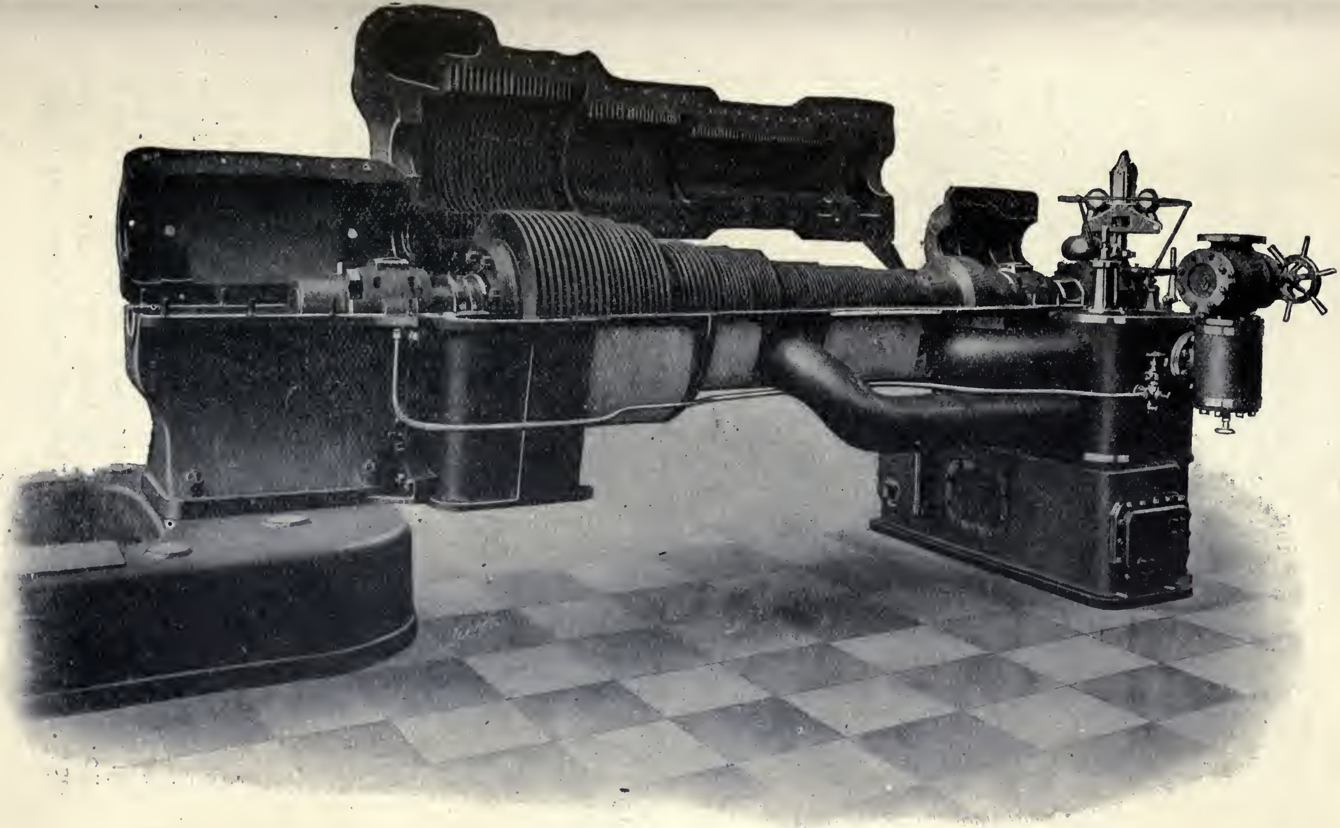


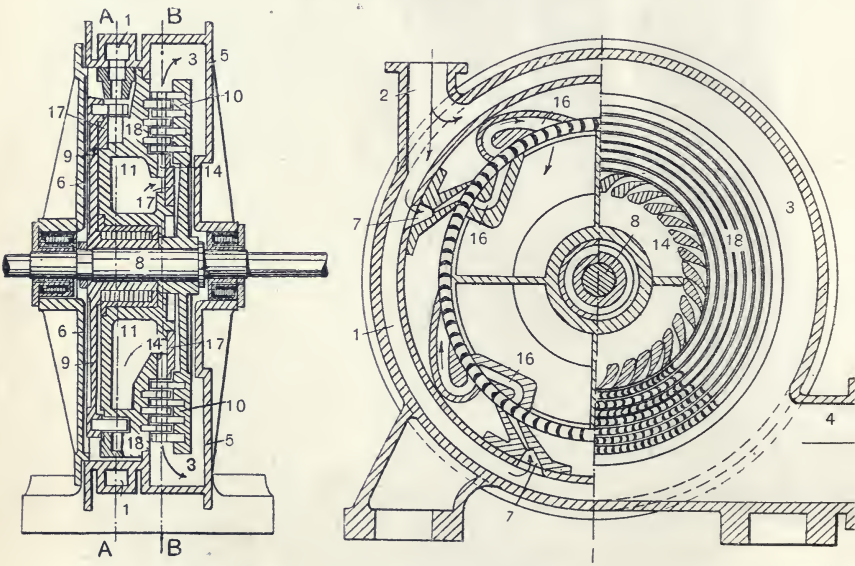
Fig. 60.—Willans & Robinson Turbine Opened Up for Inspection.

length. The hogging of the turbine casing and the distortion of the originally circular cross-section into a cross-section which if not exactly elliptical is certainly not quite circular, are especially serious in large slow-speed marine turbines.

Kolb Turbine.—This turbine will serve to illustrate what is known as the "in and out" class of turbine. It is so-called because the steam passes through the buckets or blades of the same wheel or wheels several times between stop valve and exhaust.

Some forms of the Riedler-Stumpf turbine belong to this class.

Figs. 61 and 62 illustrate the general arrangement of the turbine. The flow of steam takes place—in the



Longitudinal Section. Section on Line A A. Section on Line B B.
Fig. 61.—Kolb Steam Turbine.

main—in a radial direction. There are two sets of expansion nozzles marked 7 and 14 in Fig. 61. Referring to Fig. 62 the steam is expanded in nozzles 15 and is directed on to the moving blades. It leaves these and enters the guide passage 16, which redirects it on to the wheel at a point further round the periphery. This latter process is repeated, and then the steam is again expanded in the

nozzles 14 and flows in an outwardly radial direction through a series of alternate moving and fixed blades, projecting in concentric rings from the face of a plane disc. It is only the first expansion stage which is of the "in and out" type.

The objection to this type of turbine is that there is very considerable loss by friction in the guides 16. It is unsuitable for very large powers because of the difficulty of finding space on the periphery of the wheel for all the nozzles and guides.

These turbines must always be of the impulse type. If there were expansion in the guides 16 the temperature of the steam striking the same set of blades would be

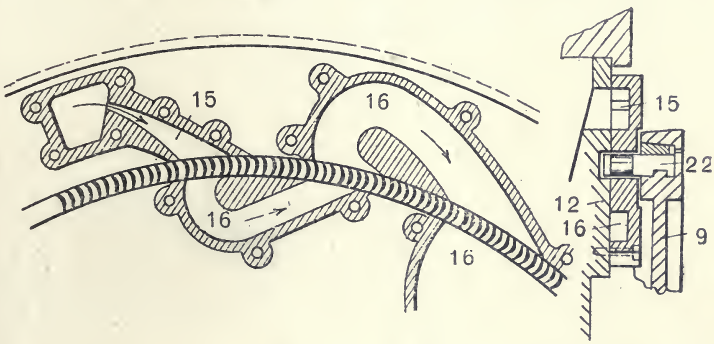


FIG. 62.—KOLB STEAM TURBINE.
Showing arrangement of Expansion Nozzles and Conducting Passages.

variable at different points round the periphery and hence initial condensation would be produced, and be accompanied by increased frictional losses in the steam. For constructive reasons it is generally better to keep the blades radial and place the steam nozzles and guides on the two sides of the wheel.

Chapter III.

STEAM NOZZLES: DISC FRICTION.

Energy of Steam.—The energy in steam consists of heat energy, pressure or potential energy, and kinetic energy. We can increase the kinetic energy at the expense of the other two, or we can partially reconvert the kinetic energy into pressure energy.

Orifices and Nozzles.—Neglecting losses, the expansion in an orifice or nozzle follows the ordinary adiabatic law.

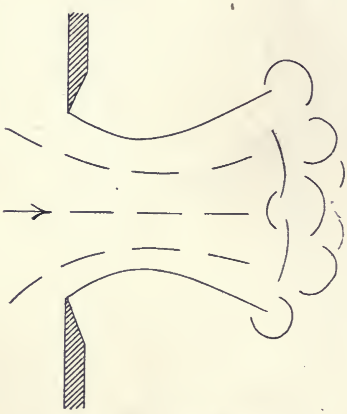


Fig. 63.—Flow from Orifice.

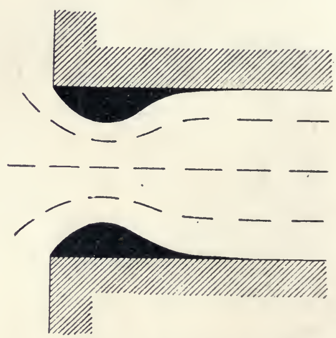


Fig. 64.—Flow in Tube Nozzle.

Pressure, volume, temperature, and dryness change just as they would do in the cylinder of an ideal engine. The work that is done during this expansion is spent in generating kinetic energy in the steam, so that this kinetic energy is readily calculable. In practice there are losses partly due to radiation, but mainly due to the friction of the steam. These losses can be kept small by suitably proportioning the nozzle and finishing its walls properly. These losses can be easily allowed for when designing a nozzle.

When steam issues from a vessel at a high pressure into a region of low pressure through a thin-lipped orifice the form of the jet is somewhat as in Fig. 63. The elementary streams of steam converging towards the orifice cause a contraction in the section of the jet just outside the orifice. A similar action occurs when using a tube nozzle. The contraction of section will still occur, so that with a sharp-edged inlet the tube at the contracted section will be partly filled with eddies (shown in black, Fig. 64). For this reason the inlet should be rounded. The pressure at the contracted section is not that of the receiving medium unless the latter is, roughly, half that of the steam before passing to the orifice. Hence

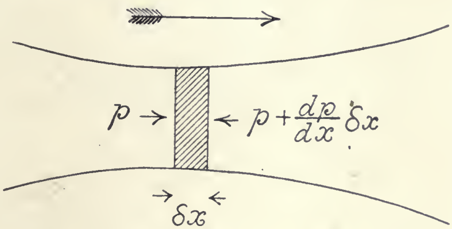


FIG. 65.—FLOW OF STEAM.

after passing the contracted section the velocity will continue to increase and the steam to expand down to the pressure of the receiving medium.

Let v = velocity in feet per second.
p = pressure in pounds per square foot.
u = volume of 1lb. of fluid in cubic feet.
$g\,\rho$ = density in pounds per cubic feet (g = 32·2).
A = area of cross section in square feet.
t = time in seconds.
M = pounds per second passing any cross section.
W = work done per pound of fluid in foot pounds.

Consider the adiabatic flow of an elementary stream of steam. Consider the forces on the two faces of a given mass of steam enclosed by two cross sections distant δx apart. The pressure intensities on the two faces are p and

$$p + \left(\frac{dp}{dx}\right)\delta x. \quad \text{(See Fig. 65.)}$$

The volume enclosed between these faces is $A\,\delta\,x$, and its mass $\rho\,A\,\delta\,x$.

Then we know that this mass multiplied by its acceleration is equal to the force producing the acceleration; that is

$$\rho\,A\,\delta\,x\left(\frac{d^2x}{d\,t^2}\right) = -\,A\left(\frac{d\,p}{d\,x}\right)\delta\,x$$

therefore $$\frac{d^2x}{d\,t^2} = -\,\frac{1}{\rho}\,\frac{d\,p}{d\,x}$$

integrating, we have

$$\frac{1}{2}\left(\frac{d\,x}{d\,t}\right)^2 = -\int\frac{1}{\rho}\,\frac{d\,p}{d\,x}\,.\,\frac{d\,x}{d\,t}\,.\,d\,t\,.\,+\text{constant},$$

$$= -\int\frac{1}{\rho}\,d\,p + \text{constant},$$

$$\text{but } \frac{d\,x}{d\,t} = v \text{ and } \frac{1}{g\,\rho} = u,$$

$$\text{therefore } \frac{v^2}{2\,g} = -\int u\,d\,p + \text{constant}.$$

Now, $-\int u\,d\,p$ is the available work in the steam and $\left(\frac{v^2}{2\,g}\right)$ is the kinetic energy generated, hence we see that these are equal. If the equation to the expansion curve is $p\,u^n$ = constant, or its equivalent $p^m\,u = C$ (a constant) where m is the reciprocal of n, then we know that between the pressures indicated by the suffixes 1 and 2, and assuming that the initial velocity is zero, we have

$$\frac{v^2}{2\,g} = W = \frac{p_1\,u_1 - p_2\,u_2}{1 - m} \quad \text{(See Fig. 66.)}$$

$$= C\left(\frac{p_1^{\,1-m} - p_2^{\,1-m}}{1-m}\right)$$

This equation can be written in other forms for use in special cases.

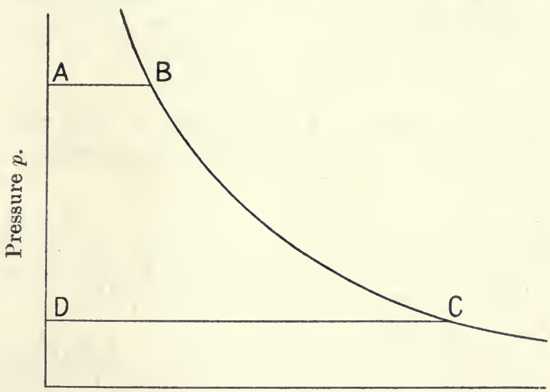
The author has determined the values of n for different

initial conditions of the steam.* The results are embodied in the following table :—

TABLE IV.

Dryness.	Pressure, Pounds per Square Inch (absolute).					
	250	200	150	100	50	20
1·0	1·129	1·133	1·136	1·139	1·143	1·145
0·9	1·120	1·122	1·125	1·127	1·130	1·132
0·8	1·109	1·111	1·113	1·115	1·117	1·118
0·7	1·099	1·101	1·102	1·103	1·104	1·105

The properties of superheated steam are not known with any certainty. Using Prof. Callendar's results, the author constructed an entropy diagram from which



Volume u.

FIG. 66.—PRESSURE VOLUME DIAGRAM.
KINETIC ENERGY GENERATED IS EQUAL TO AREA A B C D.

a value of n equal to 1·28 approximately was found. This must not be relied on more than necessary.

For adiabatic flow the nozzle should have the same relative cross sections as one of the elementary streams, and in order to reduce skin friction the cross section

* "The Engineer," April 10th, 1903: "Curves showing the Adiabatic Expansion of Steam."

should be circular. For reasons explained when considering the De Laval turbine it is desirable to make the final section of the nozzle square. The quantity of steam (including moisture) passing all cross sections is the same, hence

$$A = \frac{M\,u}{v} = \frac{M\,C}{v\,p^m}$$

and

$$\frac{A}{A_0} = \frac{v_0}{v}\left(\frac{p_0}{p}\right)^m$$

where the suffix *o* denotes the least cross section. Now A is a minimum (and equal to A_0) when $v_0\,p_0^{\,m}$ is a maximum; that is, when $(v_0\,p_0^{\,m})^2$ is a maximum. Now

$$(v_0\,p_0^{\,m})^2 = \frac{C\,2\,g\,p_0^{\,2m}\,(p_1^{\,1-m} - p_0^{\,1-m})}{1-m}$$

Differentiating with respect to p_0 and equating to zero, we see that at the least cross section we have

$$\frac{p_0}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} = \left(\frac{2\,m}{1+m}\right)^{\frac{1}{1-m}}$$

The following table gives the value of this ratio for various values of n; p is the pressure at the nozzle inlet.

TABLE V.

n	1·10	1·11	1·115	1·12	1·133	1·145	1·15	1·28
$\left(\frac{p_0}{p_1}\right)$	·584	·582	·581	·580	·578	·575	·574	·549

As an example, suppose the nozzle to work between absolute pressures (in pounds per square inch) of 215 and 1. The steam being initially dry $n = 1{\cdot}133$ and $\left(\frac{p_0}{p_1}\right) = {\cdot}578$, so that $p_0 = 125{\cdot}3$.

Hence also $v_0 = 1{,}490$ft. per second, and v_2 (the velocity at outlet) is 4,030.

Hence

$$\frac{A_2}{A_0} = \frac{1490}{4030}\left(\frac{125\cdot3}{1}\right)^{\frac{1}{1\cdot133}} = 26{\cdot}3$$

That is, the final area of the nozzle cross section is 26·3 times the least. If the nozzle had delivered at 15lbs.

absolute pressure the final velocity would be 3,190ft. per second, and the ratio of the areas only 3·03. If the nozzle passes 535lbs. of steam (initially dry) per hour the diameter at the least section must be 0·25in. This is obtained from the formula

$$A = \frac{M u}{v}.$$

It is of great importance to correctly determine the least cross section of the nozzle.

In the appendix is given a table showing the velocity at the neck with various initial pressures and conditions of the steam. The discharge in pounds per square inch (of this least cross-section) is also given. The velocity at this point is usually assumed to lie between 1,475ft. and 1,500ft. per second for practical calculations.

From the formulæ for the kinetic energy generated during an expansion, and for the ratio of the initial pressure to the pressure at the least cross-section, it follows at once that the kinetic energy generated up to the least section is

$$W_0 = \frac{p_1 u_1}{1 + m}$$

$$= \frac{v_0^2}{2 g}$$

From this the velocity at the neck can readily be calculated (see appendix). It should, of course, be remembered that the pressure is in pounds per square foot.

Referring to the least section, the discharge in pounds per square foot is

$$\frac{v_0}{u_0} = \frac{\sqrt{W_0 \, 2 \, g}}{u_1 \left(\frac{p_1}{p_0}\right)^m}$$

These formulæ hold whatever may be the dryness of the initial steam, provided only that the right value of the index m be taken. The initial volume is, of course, the actual volume per pound (including moisture).

Nozzles for Superheated Steam.—The initial pressure of the superheated steam we will denote by p_3, and the curve of expansion by $p^x u$ = a constant. The pressure

at which the steam becomes saturated (which is readily obtained from the entropy diagram) is p_1, and hence

$$\frac{v^2}{2g} = \frac{p_3 u_3 - p_1 u_1}{1-x} + \frac{p_1 u_1 - p u}{1-m} = K + W$$

W is, of course, zero if saturation has not been reached.

If the steam is still superheated at the least section the ratio of p_o to p_3 is ·552, as we have just seen. If the

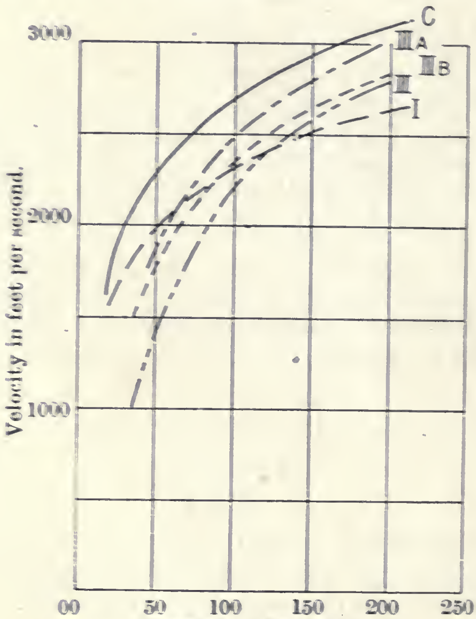


FIG. 67.—ROSENHAIN'S RESULTS.

steam has passed into the saturated state then at the least cross-section we have ($x = 1·28$)

$$\frac{2m(1-m)}{1+m} p_o^{m-1} \left(K + \frac{p_1 u_1}{1-m}\right) - C = 0$$

From this we can determine p_o.

Experimental Results.—In practice part of the kinetic energy is destroyed by friction, and reappears as heat. There is also some slight radiation loss. Apart from these losses the truth of the assumptions made in our calculations is amply demonstrated by the experiments of Stodola,* Rosenhain, and others. Fig. 67 illustrates

* "The Steam Turbine," by A. Stodola: Constable & Co., London.

some of Rosenhain's results. Curve *C* shows the theoretical velocities attained; curve I. those with a thin-lipped orifice; curves III., IIIA, and IIIB., the results with a diverging nozzle provided with a rounded inlet. Curve III. was obtained with the original nozzle; curve IIIA., when half the diverging portion had been cut away; curve IIIB., when still more of the nozzle had been removed. The nozzles discharged into the atmosphere. At low (initial) pressures the thin-lipped orifice gives the best results owing to its small frictional loss. As the pressure increases the effect of the incomplete expansion produced by such an orifice makes itself felt, and the diverging nozzle gives better results. Cutting this diverging nozzle in two so reduced the friction as to considerably improve its efficiency. A still further shortening of the nozzle, however, brings in a considerable loss due to incomplete expansion. With the nozzle IIIA., an efficiency of 94 per cent. was obtained at 200lbs. initial pressure. This diverging nozzle was the same at all pressures, and hence for low pressures it was necessarily inefficient. It would be interesting to try the effect of a thin-lipped orifice in conjunction with a turbine wheel for low pressures. The orifice would have to be placed somewhat back from the wheel to allow for the expansion outside the orifice itself.

Prof. Stodola measured the pressures in steam nozzles. He inserted a small tube connected to a manometer, along the axis of the nozzle. His results show that the expansion in a nozzle is the same as that calculated by the formula previously given, provided account be taken of the friction loss. This he estimates at from 5 to 15 per cent. of the total available kinetic energy. This loss is greater the longer the nozzle, provided that in no case is too rapid a divergence used. As to what constitutes a too-rapid divergence we are in the dark, but probably very rapid divergence could be used if the end of the nozzle was made parallel for a short distance to insure parallelism in the final jet. Then, too, we must remember that the measuring tube down the axis of the nozzle which he used would be responsible for some loss. Using a nozzle 6·3in. long, with least and greatest diameters 0·5in. and 1·45in., a measuring tube about 0·02in. diam.,

and pressure limits of 150lbs. and about 3lbs. per square inch absolute, Stodola obtained a loss of 10 per cent.

Delaporte used a nozzle 1·97in. long, and obtained a loss when discharging into the atmosphere of 5·2 per cent. Lewicki under similar conditions, but using slightly superheated steam, obtained a loss of 8 per cent. The above losses refer, of course, to kinetic energy; the velocity loss is about half this value.

Calculation Assuming Friction Loss.—Referring to the entropy diagram (Fig. 68), assume that at any point on the expansion there has been a loss of, say 8 per cent.

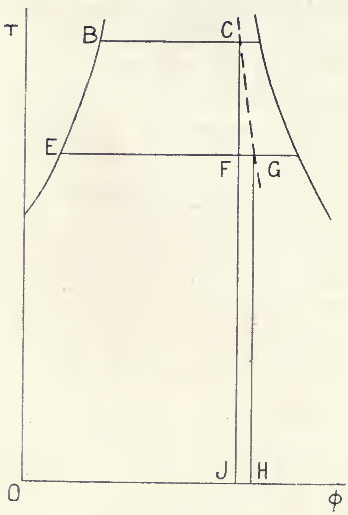


FIG. 68.

That is to say that at all points the kinetic energy present is equal to 92 per cent. of the theoretically available. *C G* is a portion of the expansion line between the pressures *B C* and *E F G*. The area *B C F E* represents the theoretically available kinetic energy (in terms of heat units) between these pressures. Then the area *C G H J* must be equal to 8 per cent. of the area *B C F E*. In this way we can readily determine the expansion curve *C G*. For constructional purposes we should take *B C* and *E F* fairly close together. Then the actual kinetic energy added between the pressures *B C* and *E F* is represented by 92 per cent. of the area *B C F E*. From

this we can calculate the velocities, and the curve $C\ G$ will give us the steam volumes, pressures, and drynesses. It is doubtful whether much loss takes place before the least section is reached, and we might take expansion as adiabatic down to that point followed by an 8 per cent. loss, or even less. Fig. 69 illustrates the conditions in a nozzle operating between absolute pressures of 215lbs. and 1lb. per square inch, with a final energy loss of 8 per cent. Steam initially dry.

Nozzle problems can be very conveniently studied by means of the author's heat diagram, which will be explained later.†

Referring to Fig. 70 (not to scale), A is the initial point for the steam before entering the nozzle. $A\ B\ D$

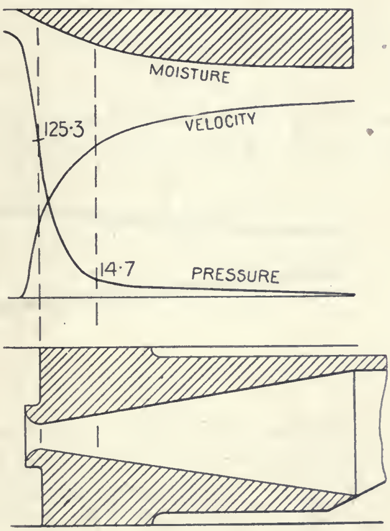


FIG. 69.—STEAM NOZZLE CONDITIONS.

is an adiabatic, $A\ E$ a throttling line. Assume adiabatic expansion down to the pressure at the least cross-section; B represents this pressure. If for an 8 per cent. loss make $D\ C$ equal to 8 per cent. of $D\ E$; join $B\ C$. Then $A\ B\ C$ is the expansion line. The method of determining the

† See Chapter XIII.

velocities and volumes for the steam at the different pressures will be fully explained later.

Improperly Formed Nozzles.—Stodola has shown experimentally that if a nozzle designed for a large pressure

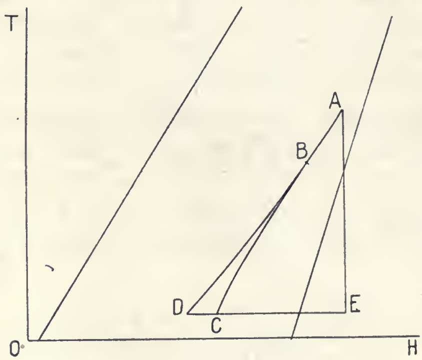
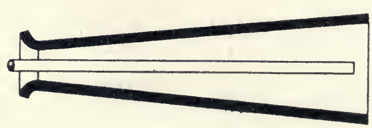


FIG. 70.

drop be used with a small pressure drop, the pressure at an intermediate part of the nozzle will be less than the



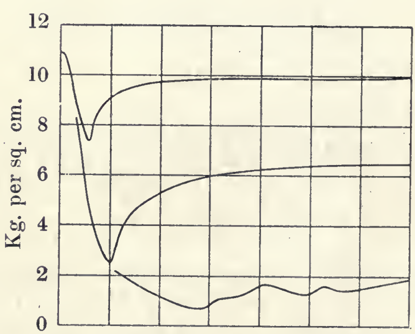


FIG. 71.—EFFECT OF BACK PRESSURE.

final pressure. Fig. 71 illustrates some of Stodola's results, and gives a sketch of the nozzle with the measur-

ing tube. We see that there takes place first an expansion just as with a big pressure drop ; followed by a compression. Now we know that in all fluids, whereas a drop in pressure may be accompanied by only a small loss, yet the reconversion of kinetic into pressure energy is always a source of very considerable loss. Clearly, then, we don't want to make our nozzle too big at the outlet.

We can represent what occurs on our heat diagram (Fig. 72). *A B C* is the expansion curve ; *C D* the com-

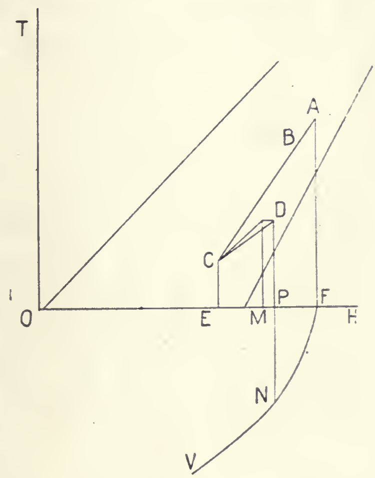


FIG. 72.

pression curve. The slope of the latter is found thus. Suppose a loss during compression of 20 per cent. Imagine an adiabatic through *C*. *E F* represents the heat converted into kinetic energy between *A* and *C*. If the compression were adiabatic *M F* would represent the kinetic energy at *D* ; but there has been a loss of 20 per cent., hence make *M P* equal to 20 per cent. of *M E* where *P* is the projection of *D*. Then *C D* is the compression curve and *P N* the final velocity. But we do not know where *C* is ; still we may get approximately the direction of *C D* in this way. Having done this we must obtain *D* by trial, such that the velocity there multiplied by the final area of the nozzle and divided by

the volume of 1lb. of steam at D gives the weight of steam flowing. Then draw D C at the required inclination, and we have C, the lowest pressure reached. The velocity at C is obtained by projecting on to the velocity curve F N V. It is clear from the diagram that a slight variation in the initial conditions will considerably affect the position of C. This was found to be so in Stodola's experiments. Lindmark makes use of a diverging nozzle to compress the steam leaving the wheels of a turbine, previous to its reconversion into kinetic energy in a succeeding set of nozzles.

Stodola also experimented with a diverging nozzle to the end of which was coupled first a converging nozzle,

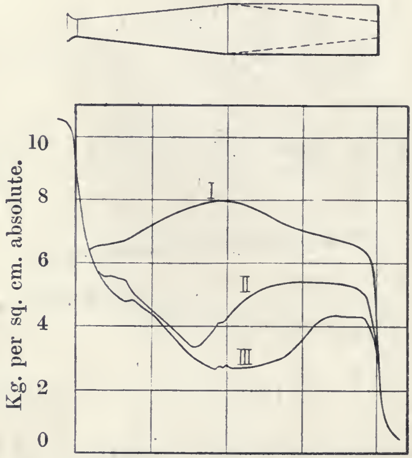


FIG. 73.—RECONVERGING NOZZLE.

and finally a cylindrical tube. With the converging ends he obtained curves somewhat like curve I. in Fig. 73. As this converging end was made more and more like a cylinder the curve approached that shown in curve III.; curve II. being for a slight convergence of the end nozzle. At first sight it seems to be impossible for us to obtain an increasing pressure in a cylindrical end to a nozzle; friction, it would seem, should require a positive pressure to overcome it, not a negative pressure. The explanation of the phenomena seems to be somewhat as follows. As the pressure in a nozzle falls we have an increase in

both the volume and velocity. Now the quantity of steam passing all cross-sections is $\frac{v\,A}{u}$ and is constant.

The ratio $\frac{v}{u}$ increases with a fall of pressure down to about 0·6 of the initial pressure, the exact value of this ratio depending on the losses. Below this pressure a further decrease in pressure is accompanied by a decrease in the ratio $\frac{v}{u}$. Consider the cylindrical portion of the nozzle ; evidently $\frac{v}{u}$ must be constant. But the effect of friction is to decrease v and increase u, and to balance this effect we must have at the same time an independent increase in the ratio $\frac{v}{u}$.

For pressures approximately greater than 0·6 of the initial pressure this can only be attained by a fall in pressure, whereas for pressures below 0·6 of the initial pressure there must be an increase in pressure to counterbalance the effect of friction, so that we get, as in curve III., a rising pressure in the cylindrical end to the nozzle. Where the nozzle is tapered the pressure curve will depend on the rapidity of the taper ; a slight taper will approximate nearly to the same conditions as a cylindrical nozzle, the decrease in the cross-section (A) causing (curve II.) a still greater rise in pressure. In the case represented by curve I. the pressure at the beginning of the converging portion is considerably more than 0·6 of the initial pressure, and hence the decrease in the cross-section and the decrease in the ratio of v to u due to the results of friction, must be balanced by a decrease in the pressure.

Stodola also measured the pressures along the axis of flow of plate orifices. Fig. 74 illustrates some of his results when using a short tube orifice with a sharp inlet. It will be seen that where the outlet pressure is less than that which would occur at the least section of a properly-formed nozzle (about half the initial pressure) vibrations occur as the steam leaves the orifice. The expansion in the orifice has not been complete, so that when the steam leaves the orifice there is a sudden expansion which, like the sudden expansion of the gases leaving a gun barrel,

gives rise to vibrations which are by the manner of their formation identical with sound vibrations. These vibrations or fluctuations of pressure are stationary in space. Similar fluctuations in pressure have been observed by other experimenters. They are a source of loss, and can be avoided by using properly-formed nozzles instead of orifices.

The effect of the sharp inlet is seen in curve A, where the pressure drops at entrance to about 3·3 kilogrammes per square centimetre, then rises to about 4·4, and after a few oscillations leaves the nozzle, dropping to the vacuum pressure as it does so. With a rounded inlet

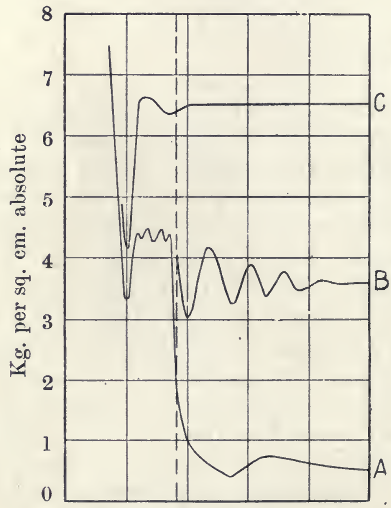


FIG. 74.—SHARP-EDGED PLATE ORIFICE.

no vibrations occur until the steam leaves the orifice. The vibrations which occur outside the orifice are well illustrated by curve B.

In designing a steam nozzle attention should be paid to the following points :—

(1) The nozzle should be designed for *average* working conditions.

(2) The surface of the nozzle should be as small as possible consistent with good proportions.

(3) The inlet should be well rounded—not too large a curve—and should project back into the main body of the steam.

(4) The nozzle should be designed for a loss of from 4 to 10 per cent., according to operating conditions.

(5) If the divergence of the nozzle is rapid the exit should be parallel for a short distance, say about a fifth of the diameter.

(6) The internal surfaces to be polished.

The nozzle may be made of gun metal, although nickel steel is preferable for use with highly-superheated steam.

Best Taper for Nozzles.—The cross-sections of the nozzle at the neck and at outlet are determined by the pressures and temperatures between which the nozzle has to work. The longer the nozzle the smaller the taper, and the less will be the loss of kinetic energy by spreading. For instance, if a particle of steam is moving at an inclination of 45° to the nozzle axis, then only half the kinetic energy is available in the direction of the axis, unless some deflecting arrangement is used. The other half is lost by spreading. The simplest deflecting arrangement will consist of a short parallel section at the outlet end of the nozzle.

On the other hand, the shorter the nozzle the smaller the friction, the friction being just proportional to the length.

These two losses are illustrated in Fig. 75, in which also the dotted curve represents their sum. It will be seen that the total loss is a minimum at a certain particular taper. In calculating the loss of kinetic energy by spreading it must be remembered that the average inclination of the moving steam to the axis is not that of the walls of the nozzle, but is something less, the actual value depending on the ratio of the two cross-sections to the length—the rounded inlet to the nozzle can usually be neglected so far as the length is concerned. The greater the capacity of the nozzle and the smaller the pressure drop, the smaller the average inclination of the steam for a given slope of the walls. A nozzle of large capacity can therefore have a somewhat more rapid taper than a small nozzle.

As the co-efficient of friction is influenced by the state of the walls, the dryness or amount of superheat in the steam and the density, it is very difficult even to make an estimate of the best taper. From some rough calculations made by the author a taper of from 1 in 15 to 1 in 20 for a small nozzle with a large pressure drop seemed to be best, but the figures are not of much value.

In actual practice this question of the best taper is not so important as might appear from a consideration of the nozzle by itself. When in use, the fact that some of the steam is not moving in a perfectly axial direction at the nozzle outlet does not mean that the component of the kinetic energy perpendicular to the nozzle axis is lost.

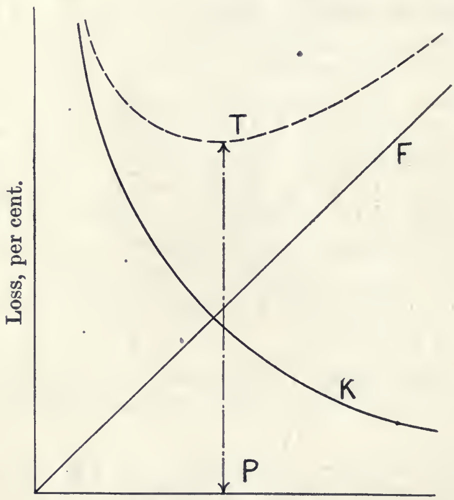


FIG. 75.—BEST TAPER FOR NOZZLES.
K, Kinetic loss.
F, Friction loss.
T, Total loss.

Some of it is converted into work by the moving blades, so that the actual loss from spreading will not be very large. It is worth noting in this connection that some turbine builders who have tried nozzles with and without a parallel end-piece find no appreciable gain due to the use of the parallel end under working conditions, suggesting that the effect of the spreading of the nozzle is minimised by the action of the moving blades in catching and deflecting *all* the steam leaving the nozzles.

In view of this fact, it would seem that friction is the more important source of loss, and that comparatively rapid tapers are advisable.

In De Laval turbines the taper of the nozzle depends very much on the pressure drop, as nozzles of very different proportions have to fit the same turbine casing.

Friction of Turbine Discs.—The friction of a turbine disc rotating in a bath of steam or gas may be divided into two parts : (1) there is the skin friction, and (2) the fan or eddy loss due to the churning action of the blades. The latter is the more serious of the two. The frictional resistance is proportional to the square of the velocity

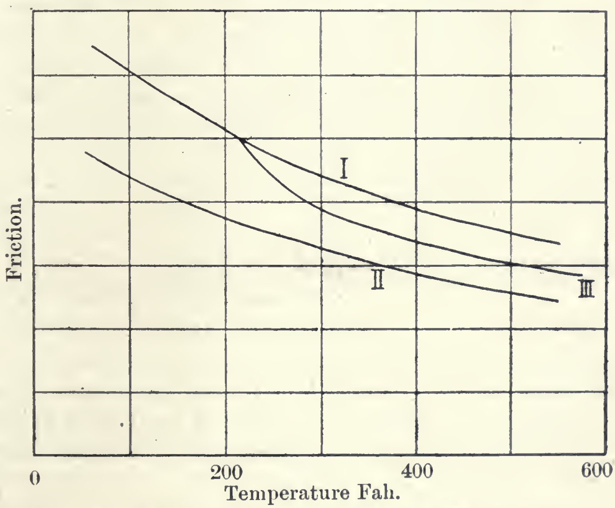


FIG. 76.—DISC FRICTION : EFFECT OF MOISTURE.

in both cases, except at low velocities and near the centre of the disc, when the skin resistance is only proportional to the velocity. Neglecting this modifying influence the total frictional resistance will be proportional to the square of the velocity, and the frictional loss (in work units) will be proportional to the cube of the velocity. Experiments by Stodola, Lewicki, and others conclusively prove this to be true, except that the index of the velocity seems to be about 2·9 and not exactly 3. Since the churning action of the blades is the more serious of the

two components of the frictional loss, it is not surprising that the loss is greater when the concave faces of the blades advance, being about five or six times as large as for forward running (convex face advancing) when the wheel has a large space or bath to run in, but is only about 1·2 times as great when the blades are surrounded by a casing with little clearance, for in this case the blades do not do so much churning but rather carry the steam round bodily.

This disc friction is proportional to the density of the surrounding medium, and is greater in air than saturated steam, and still more so than in superheated steam. The following table gives Lewicki's results .—

TABLE VI.

Medium.	Temperature F.	Wheel Friction in H.P. at	
		14·7lbs. per sq. in.	5·3lbs. per sq. in.
Air	86	4·5	..
Saturated steam ..	212	3·25	1·47
	253	2·81	0·938
Superheated steam ..	363	2·22	..
	471	2·02	..
	572	1·85	0·5916

From the table it appears that when allowance has been made for the relative densities of air and saturated steam the steam offers more resistance to the wheel than does the air; and that unless there is an abnormal expansion in the early stages of superheating, the friction of superheated steam is—after allowing for the effect of the different densities—less than for saturated steam. This would seem to prove that the moisture in the steam is the cause of much of the friction, and hence the desirability of using dry or superheated steam where possible.

In order to illustrate this point, Fig. 76 has been drawn from Lewicki's results. In the construction of the curves it has been assumed that the friction is proportional to the density, and that the density is inversely proportional to the absolute temperature. In order to bring the results

for air and steam to a common basis, they have been corrected for the known difference in density at 212° Fah. Curve I. shows the results for saturated steam based on the result obtained at 212° Fah. Curve II. shows the results for air. Curves I. and II. are, of course, rectangular hyperbolas. Curve III. shows the actual results obtained with superheated steam. All three curves are, as previously mentioned, reduced to a common density at 212° Fah.

Now, assuming that the density does vary inversely as the absolute temperature, any differences in the frictional resistances offered by air, saturated steam, and superheated steam must be due to some difference between the natures of the fluids; such as the presence of moisture. We see that as the superheat increases (Curve III.) the friction gets less, but never falls as low as for air. We also note that in the early stages of superheating there is a more than proportionate decrease in the friction, suggesting that there may possibly be some moisture present. These curves suggest that the presence of moisture increases the friction.

To reduce disc friction aim at the following objects:—

(1) Low steam pressure.
(2) A fair value of superheat.
(3) Small clearances between blades and casing.
(4) Low peripheral velocities.
(5) Small surface areas.
(6) Large clearances between disc and casing.

Chapter IV.

BLADES.

Blades.—The following points should be kept in mind when deciding the blade forms :—

(1) The blade angles should be such that the steam is received by the fixed or moving blades without shock. That is, we require the direction of motion of the steam relative to the blade to be parallel to the face of the blade.

(2) The ratio of the friction loss to the work done in the moving blades to be as small as possible.

(3) Leakage through the clearance spaces between the rotating and stationary parts to be as small as possible.

(4) The perimeter of the flowing belt or stream of steam to be as small as possible for a given cross-section.

(5) An unsymmetrical moving blade in an impulse turbine gives rise to end thrust.

(6) The loss to exhaust should be as small as possible.

(7) The blade must be strong enough.

(8) The passage cross-sections between the blades must be correctly proportioned.

The first condition, that the steam enter the blades without shock, is secured by a little consideration of the velocity diagrams to be discussed shortly. In many-stage turbines, and especially in reaction turbines of large size, the second condition (that the friction loss be a minimum) is, from the point of view of efficiency, probably the most important. In order to secure it we must adopt certain blade angles, not, as a rule, those which keep the steam velocities at a minimum, because in doing so we generally have to so increase the size of the turbine passages, and frequently also the length of the rotor, that the increased surface exposed to the steam increases the friction more than the reduction in velocity decreases it. The clearance leakage is

necessarily greater in a small than in a large turbine, and is less the smaller the velocity of the steam in a direction parallel to the shaft, so that in very small turbines where the clearance leakage is large it may be desirable to reduce it at the expense of the friction loss by decreasing the axial steam velocity or by restricting the circumferential length of the steam path. Unless the axial clearances are very small this partial peripheral admission in which the fixed blades only cover a portion of the periphery leads to rather serious losses, especially so in reaction turbines, so that in turbines of the Parsons type it is common practice to have full peripheral admission at all parts of the turbine and to use fairly-large axial clearances between the fixed and moving blades. These clearances are made as large as 1in. in some large turbines at the low-pressure end, but this is exceptional; from $\frac{1}{10}$in. to $\frac{1}{3}$in. being more usual, the latter figure being the clearance at the low-pressure end. Provided that it does not necessitate any considerable increase in the cost of manufacture or increase the risk of injury to the blades, the smaller these axial clearances the better, although the efficiency is not appreciably affected by small variations in these clearances. In impulse turbines it is, however, usual to have partial peripheral admission at the high-pressure end where the volume of the steam is small. In order, then, to reduce the external surface of the flowing belt of steam the nozzles or fixed-blade openings should be grouped together into a few symmetrically-placed groups. Thus we saw in Fig. 28 that the nozzles for the fourth stage of a Curtis turbine were grouped into four sets. It is not advisable to have these groups too widely spaced, as that might give rise to an inequality of temperature distribution, and cause a distortion of the casing. The practical value of this grouping of the nozzles as contrasted with the old method of wide and equal spacing has been conclusively demonstrated by tests on turbines of the De Laval type, the eddy losses at the boundaries of the many streams of steam in the latter case being markedly greater than in the former.

The loss to exhaust is of most importance in single-stage turbines, such as the De Laval and some forms of the Riedler-Stumpf, and in these cases is the most

important condition in determining the blade angles. It is also of considerable importance in Curtis turbines, for here there is practically no attempt made to prevent the loss by delivering the exhaust steam from one stage (not from one wheel into the adjacent guide of the same stage) into the nozzles of the next stage. This is so because the large pressure difference between consecutive stages makes it practically necessary to dish the intermediate diaphragm to a very considerable extent. This was clearly illustrated in Fig. 26. In many-stage turbines, particularly those of the reaction type, this loss to exhaust should be small, the exhaust from one stage being received without shock or great loss in the next stage.

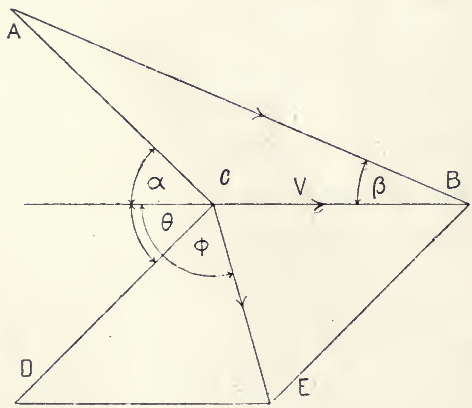


FIG. 77.—VELOCITY DIAGRAM FOR SINGLE-STAGE IMPULSE TURBINE.

It will have been noted that whereas it is important in a reaction turbine to reduce the radial clearances, and axial clearances are not of such importance, the reverse is the case with impulse turbines, radial clearances being usually quite large. This is fortunate, since the construction of an impulse turbine is not such as to allow of a stiff rotor, which is very necessary if the radial clearances are to be small.

Blade Angles: Single-stage Impulse Turbine (De Laval Type).—The angle of a blade at a certain edge is not, as a rule, either the angle of the back or front face at that edge, but it is the mean of these two angles.

In Fig. 77 we have the velocity diagram for a single-stage turbine. β is the angle the axis of the steam nozzle makes to the plane of rotation of the wheel. $A\ B$ represents the absolute velocity of the steam as it leaves the nozzle ; $C\ B$ the velocity of the blade. Then $A\ C$ must represent in magnitude and direction the velocity of the steam relative to the moving blade. Therefore the angle a must be the correct inlet angle for the blade. $A\ C$ is the direction in which a particle of steam, actually moving along $A\ B$, appears to be approaching the blade when the observer is on the moving blade itself. Unless a further expansion take place within the blades (which in an impulse turbine there is not) the relative (to the blade) velocity of the steam at outlet will be equal to that at the inlet. Actually, friction will reduce it.

Let $C\ D$ be this outlet velocity relative to the wheel. Complete the parallelogram $D\ C\ B\ E$. Then the diagonal $C\ E$ represents the absolute outlet velocity both as to magnitude and direction. For convenience we shall call $A\ B$, $A\ C$, $C\ E$, and $C\ D$ v_1, v_2, v_3, and v_4 respectively. Then we have with a wheel velocity V

work done $= \frac{V}{g}\left(v_1 \cos\beta - V + v_4 \cos\theta\right)$ per pound of steam.

The available energy at the nozzle outlet is $\frac{v_1^2}{2g}$, and the loss to exhaust is $\frac{v_3^2}{2g}$.

The remainder of the energy which has not so far been accounted for is spent in overcoming friction and (a small percentage) in radiation.

In order to avoid end thrust it is usual to make $\theta = a$, in which case if $v_4 = v_2$ the efficiency is

$$\frac{\text{Work done}}{\text{Original kinetic energy}} = \frac{\frac{2}{g}\left(v_1 \cos\beta - V\right)V}{\frac{v_1^2}{2g}}$$

$$= \frac{4\ V\ (v_1 \cos\beta - V)}{v_1^2}$$

The turning effort on the disc is $(v_1 \cos \beta - V + v_4 \cos \theta)\frac{1}{g}$

This is a maximum when $v_1 \cos B = V$. If B were 20° (the usual value for a De Laval turbine) then α and θ would both be 36° and V should be 0·47 v_1, or under ordinary conditions about 1,900ft. per second. For practical reasons this velocity is never attained, so that α will be less than 36°. Also the velocity at outlet relative to the blade will be less than the inlet velocity, because of the frictional resistance. With a symmetrical blade this would give rise to end thrust. By slightly altering the outlet angle a complete balance is obtained.

It is important in the single-stage impulse turbine to have the kinetic energy in the steam leaving the wheel

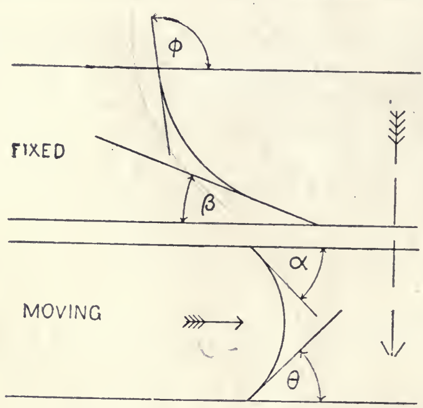


FIG. 78.—IMPULSE BLADE ANGLES.

as small as possible. In the many-stage impulse turbine this is not of much consequence, as it is possible to use the kinetic energy in the exhaust from one stage for performing useful work in the next.

Many-stage Impulse Turbine.—The velocity diagram for the moving blades is like that for the single-stage turbine. In order that the kinetic energy in the exhaust from one wheel may be utilised in the next set of nozzles the inlet angle of the nozzles should be made equal to ϕ the angle made by $C E$ to $B C$. This is illustrated in Fig. 78. We shall see later how to determine the most efficient blade angles.

Curtis Turbine.—Here we have two or more sets of moving blades for each set of nozzles. The velocity diagrams for each set of moving blades are similar to those for the single-stage impulse turbine. In the second set of wheels the inlet velocity of the steam (absolute) is the same, neglecting friction, as the absolute velocity with which it leaves the first set. The intervening set of guides is intended merely to alter the direction and not the magnitude of the velocity. Actually the velocities are considerably reduced by friction and splash or eddy losses. Fig. 79 illustrates the determination of the blade angles for a stage with two sets of moving blades. *A B* is the velocity of the steam as it leaves the nozzles; *C E* the absolute velocity with which it enters the guides; *E F* the velocity with which it leaves the guides; and *G H*

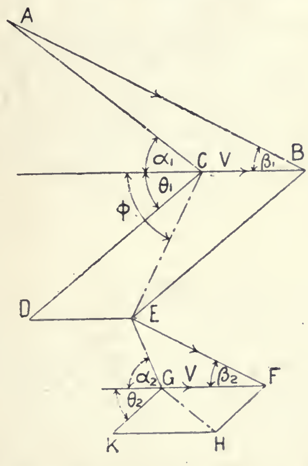


FIG. 79.—VELOCITY DIAGRAM FOR CURTIS TURBINE.

the final exhaust velocity. If there were no friction *A C* and *C D* would be equal, and so also would *E G* and *G K*.

ϕ is the inlet angle of the guides, and β_2 the outlet angle. In the Curtis turbine no attempt is made to utilise the final exit velocity *G H* in the next set of nozzles. It is usual to so arrange the velocities that the blades (with the exception of the nozzles) are all symmetrical; thus $\alpha_1 = \theta_1$ and $\alpha_2 = \theta_2$ also $\phi = \beta_2$.

The American General Electric Company has recently patented a means which is intended to reduce the losses between stages. The arrangement is illustrated in Fig. 80. A set of fixed blades 13 are placed in front of the nozzle inlets to catch and direct the steam leaving the last set of moving blades of the previous stage. Two

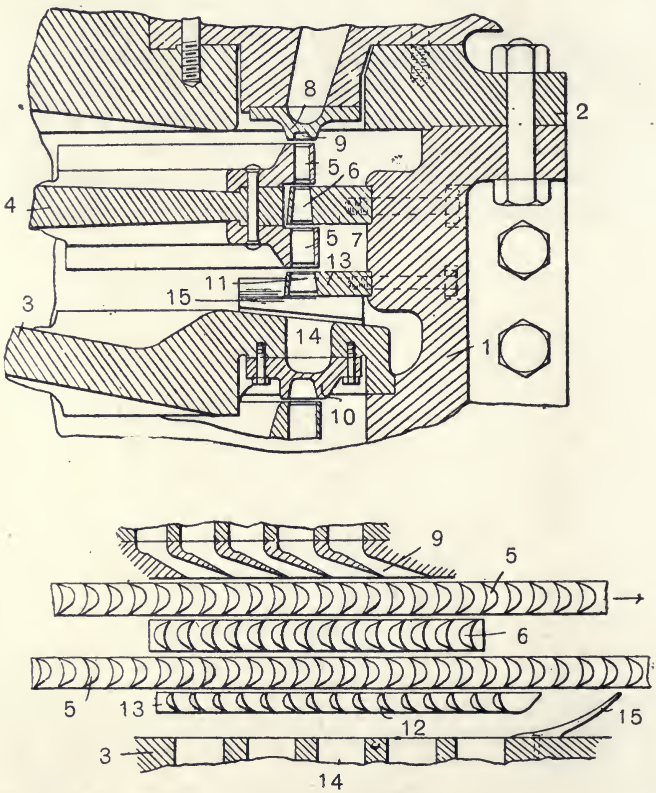


FIG. 80.
ARRANGEMENT OF TURBINE FOR PREVENTING EDDYING AND REBOUNDING OF STEAM.

deflectors or scoops 15 are also attached to the nozzle diaphragm to prevent spreading of the steam leaving the moving blades.

In another patented device for use with Curtis turbines the whole of the steam is made to pass over a barrier before entering the nozzles. The barrier will increase

the inter-stage losses, but it is hoped that the moisture in the steam will not surmount the barrier, and can be drained away.

Parsons Turbine.—In this reaction turbine kinetic energy is generated in both fixed and moving blades, and absorbed in the latter. For simplicity in construction and on account of the greater efficiency it is usual to make the stationary and moving blades of similar form. We shall take them as being so in what follows, α and β being the inlet and outlet angles respectively. Then clearly (Fig. 81) $v_1 = v_4$ and $v_2 = v_3$, and the kinetic

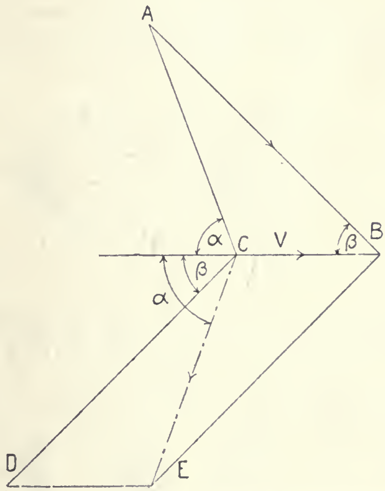


FIG. 81.—VELOCITY DIAGRAM FOR REACTION TURBINE.

energy generated in the fixed blades is the same as that generated in the moving blades, and is equal in amount to

$$\frac{v_1^2 - v_2^2}{2g}$$

Since there is supposed to be no loss when the steam leaves the moving blades, all the kinetic energy being used in succeeding stages, this is also half the work done or half the "indicated" work. There is no need then to consider the friction loss, for since the generation of kinetic energy and its loss by friction are going on simultaneously we may consider that the full

velocity is never attained (which is true), and our calculations are based entirely on actual velocities. The way in which we do allow for frictional (including eddy and leakage) losses will be explained later. This method is not strictly applicable to most of our calculations for impulse turbines, because, except in the nozzles, the frictional reduction in velocity is not coincident with the generation of kinetic energy. That is to say the velocity is first created by the free expansion and then partly lost owing to the effect of friction, whereas in the reaction turbine the velocity is not created, seeing that the generation and reduction take place simultaneously.

Referring to Fig. 81, the tangential effort or pressure of the steam per pound of steam per set of moving blades is, in pounds,

$$\frac{1}{g}\left(v_2 \cos \alpha + v_1 \cos \beta\right)$$

The work done in foot-pounds per second is

$$W = \frac{V}{g}\left(v_2 \cos \alpha + v_1 \cos \beta\right)$$
$$= \frac{v_1^2 - v_2^2}{g}$$

The work done in each set of moving blades (where the blade and steam velocities are the same) in any given section is the same. The value of V, the peripheral speed of the middle height of the blades, is known, and the other velocities can be calculated from the known blade angles, so that the "indicated" work is readily calculated.

Fig. 82 will show more clearly what the angles referred to measure. In most reaction turbines the values of the angles are not those of Fig. 82. The outlet angle usually lies between 20° and 30°, the inlet angle being not very different from a right angle, although occasionally as small as 65°. The blade profiles then become more nearly crescent-shaped, with the inlet edge somewhat blunter than the outlet edge.

Passage Cross-sections.—It is of great importance to correctly determine the areas of the passage cross-sections at different points along the turbine.

If A = area of cross-section taken perpendicular to the direction of flow, in square feet ;

v = velocity of steam (including moisture) in feet per second ;

u = volume of 1lb. of steam (including moisture) in cubic feet ;

W = pounds of steam passing through the turbine per second under maximum load conditions ;

θ = inclination of the direction of flow to the axis of rotation (shaft) ;

Then

$$W = \frac{A\,v}{u},$$

and

$$A = \frac{W\,u}{v}.$$

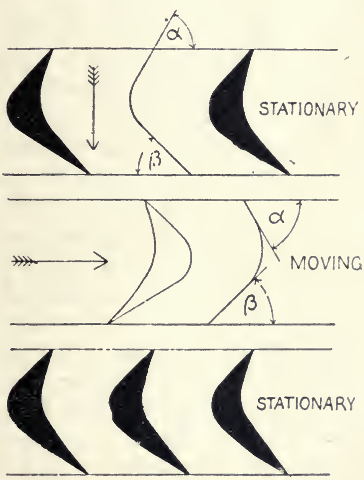


FIG. 82.—BLADES OF REACTION TURBINE.

The area of cross-section in a plane perpendicular to the shaft will be

$$\frac{A}{\cos\theta} = \frac{W\,u}{v\cos\theta}.$$

This area is measured between rotor and casing, and includes not only the area swept out by the blades, but also the clearance area between the blade tips and the casing or rotor, as the case may be.

The method of determining u, the volume of the steam, will be described later. At present we may just indicate

the method. We assume a certain efficiency for the turbine, say 60 or 65 per cent. From our velocity calculations we determine the indicated work done previous to the arrival of the steam at the particular cross-section; divide this by the efficiency and we have the maximum possible or available work between the initial and present conditions. This gives us the pressure at the cross-section under consideration. The volume of the steam is then known, because after allowing for radiation and leakage, the excess of the available over the indicated work has been spent in increasing the volume of the steam (evaporating moisture or superheating) over and above what it would have been had the expansion been adiabatic.

Number of Stages.—First fix on the number of sections (usually three in a reaction turbine) in which the diameter remains constant. Then fix the pressure drop for each section so as to give a suitable blade height (not too long at outlet nor too short at inlet). Calculate the theoretical work between the pressures decided on for the particular section under consideration; multiply this by the efficiency, and we obtain the indicated work per pound of steam for the section. Divide this by W, the indicated work per stage as calculated from the velocity diagrams, and we obtain the number of stages in that section. For instance, if the theoretical (available) work for the section is 100 B.Th.U., and the efficiency 60 per cent., with W equal to 5 B.Th.U. the number of stages in that section will be 12, each stage consisting of a row of moving and a row of stationary blades. The indicated work in each stage for any one section is the same if we neglect the change in the *mean* diameter of the blade circle in the section.

The efficiency previously mentioned is the ratio of the indicated work done by the turbine to the theoretically available work in the steam, and equals:

$$\frac{\text{B.H.P.} + \text{bearing friction} + \text{disc friction}}{\text{theoretical work in steam.}}$$

If allowance is made for radiation—which is more correct—the efficiency is somewhat greater than the above, being equal to:

$$\frac{\text{B.H.P.} + \text{bearing friction} + \text{disc friction}}{\text{theoretical work} - \text{radiation.}}$$

The efficiency varies at different points along the turbine, being in general somewhat higher toward the low-pressure end on account of the excessive leakage over the blade tips at the high-pressure end. The friction is probably somewhat greater at the high-pressure end, although we must remember that at the low-pressure end the steam is wet, the surface exposed very much greater, and the velocity higher, the density of the steam, however, being very much less. The large blade length as compared with the rotor diameter at the low-pressure end will, moreover, lead to appreciable differences in the blade speed between the rotor end and the outer end, the steam velocity remaining approximately unaltered, and thus giving rise to some eddy losses. For these reasons it does not appear that, apart from the effect of clearance leakage, the low-pressure end will be more efficient than the high-pressure end, except in small turbines. What experimental results there are bearing on this point are not very consistent.

Having allowed from 3 to 6 per cent. for radiation, according to the size and type of turbine—this loss being greater the smaller (power) the turbine and the lower its blade speeds—we assume efficiencies of from 50 to 67 per cent. at the high-pressure section, from 55 to 70 per cent. at the intermediate section, and from 60 to 73 per cent. at the low-pressure section. The lower of the values given above are for small turbines. By taking great care in the forms and arrangements of the blades, we may hope to improve on the above efficiencies.

For the purposes of quick calculation in the drawing office it is advisable to have a curve showing the indicated work per stage at all blade speeds for the standard blade angles; also a straight-line chart giving the blade speed at all diameters for different revs. per minute, and a table of theoretical works for different initial conditions. Owing to the variation in the mean diameter of the blade circle in the low-pressure—and even the high-pressure—section it is frequently desirable in our calculations to assume as many as eight different diameters in the ordinary Parsons turbine.

End Thrust.—End thrust in a reaction turbine is not, as a rule, due to the lack of symmetry in the blades but to the fall in pressure which takes place in the moving blades. In

addition there is a frictional drag on the blades, and in the Parsons turbine as usually constructed there is a slight end thrust due to the arrangement of the rows of blades into groups of equal blade height which causes a deviation of the actual from the theoretical velocities.

In order to illustrate the effect of the fall of pressure in producing end thrust, we will consider a simplified case. Referring to Fig. 83, suppose $A\,B\,C\,D$ to be a blade, and suppose the pressure on the portion ABD to be equal to p_1 and the pressure on the portion $B\,C\,D$ to be p_2. Let $A\,H$ and $C\,N$ be parallels to the direction of the fall of pressure —the shaft in most cases—then the resultant pressure

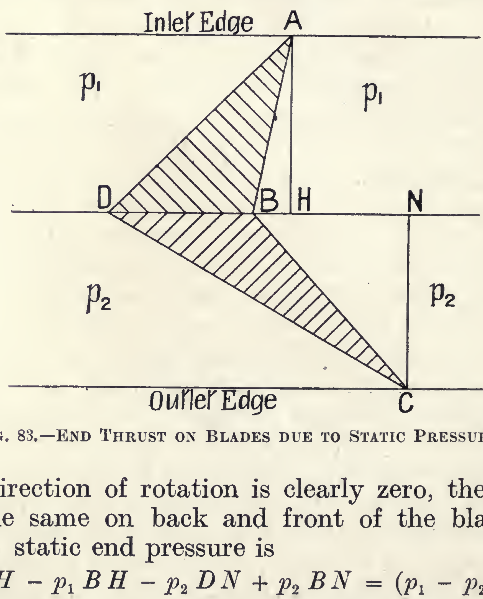


FIG. 83.—END THRUST ON BLADES DUE TO STATIC PRESSURE.

in the direction of rotation is clearly zero, the pressure being the same on back and front of the blade. The resultant static end pressure is

$$p_1\,D\,H - p_1\,B\,H - p_2\,D\,N + p_2\,B\,N = (p_1 - p_2)\,D\,B.$$

This end pressure is clearly not zero unless p_1 is equal to p_2. We see, then, that the fall of pressure in the moving blades of a reaction turbine produces end thrust, but not rotation.

In practice the conditions are not quite so simple as those chosen for the above example ; p_1 and p_2 are not uniform on the two portions of the blade, and further the reversal of the directions of slope at D and B are not always in the same plane of rotation. If, however (Fig. 84), x is the distance measured parallel to the plane

of the blade edges, $p + \delta p$ and p are the pressures at the inlet and outlet edges of the blade, and a is the circumferential pitch of the blades; we may say with reasonable accuracy that the mean pressures on the inlet and outlet halves of the blades are respectively

$$p + \frac{3\,\delta p}{4} \text{ and } p + \frac{\delta p}{4}$$

and that the end thrust will then be (per unit blade height),

$$\frac{x\,\delta p}{2}$$

The more rows of blades there are in the turbine the

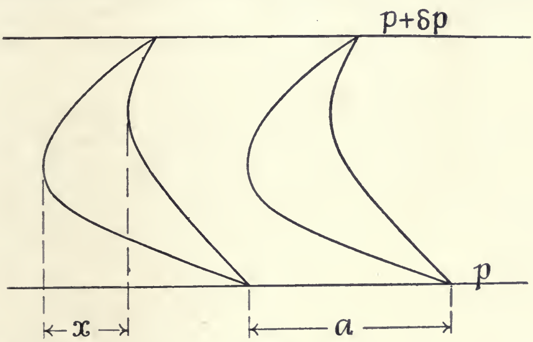


FIG. 84.

more nearly true does this become, and in all practical cases it is very nearly true.

If w = pounds of steam passing per second,
v = axial velocity of steam in feet per second,
A = annular area swept by blades,
u = volume of 1lb. of steam in cubic feet, then

$$w\,u = v\,A.$$

Further, the projected area of the blade sections in the above annular area is clearly

$$\frac{x\,A}{a} = K\,A$$
$$= \frac{K\,w\,u}{v}$$

and the end thrust in this row of blades is

$$\frac{K\,w\,u\,\delta p}{2\,v}$$

and the total end thrust for the section—constant diameter and velocity—between the pressures p_1 and p_2 is

$$\frac{1}{2}\int \frac{K w u \,\delta p}{2 v}$$

$$= \frac{K w}{4 v}\int u \, dp.$$

The reason for the factor one-half is that half the fall in pressure takes place in the fixed blades and produces no end thrust on the rotor.

Now $\int u \, dp$ is equal to the area of the diagram *A B C D*, where *B C* is a portion of the *actual expansion line*, as shown in Fig. 85.

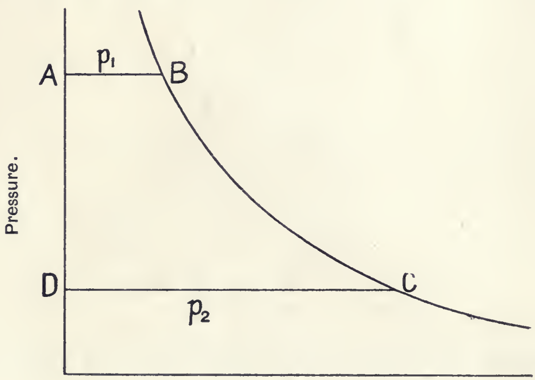


FIG. 85.

If this expansion line is of the form $p^m u = C$, then the area *A B C D* is equal to

$$\frac{p_1 u_1 - p_2 u_2}{1 - m}.$$

The end thrust is therefore

$$\frac{K w}{4 v}\left(\frac{p_1 u_1 - p_2 u_2}{1 - m}\right)$$

$$= \frac{K (p_1 A_1 - p_2 A_2)}{4 (1 - m)}$$

The values of *K* and *m* will depend on the spacing of the blades, and the initial conditions of the steam. If

we take m equal to $0 \cdot 9$ and $K = 0 \cdot 4$—common values in actual practice—we see that the end thrust is equal to

$$p_1 A_1 - p_2 A_2$$

This is the common rule used by designers in calculating the sizes of the balance pistons. In practice, allowance must be made for the value of K if this is not such as to make the thrust equal to the last expression, and also for the pressures on the steps and ends of the rotor.

It will be seen that the end thrust is inversely proportional to the axial velocity of the steam, and hence low steam and peripheral velocities, such as are used in marine work, enable the turbine to have a large end thrust with which to partially balance the propeller thrust.

We can easily estimate the friction thrust. Thus, if the friction of the steam on the walls of the blade passages amounts to $2k$ per cent., k per cent. being lost in the fixed blades and k per cent. in the moving blades, the friction loss in one set of moving blades per pound of steam is

$$\frac{k\,(v_1^2 - v_2^2)}{g}$$

Now, this loss must equal the mean frictional force (f) multiplied by the mean velocity relative to the surfaces so that we have

$$f = \frac{k\,(v_1^2 - v_2^2)}{g} \quad \frac{2}{(v_1 + v_2)}$$

$$= \frac{2\,k\,(v_1 - v_2)}{g}$$

Now, since the friction is proportional to the square of the velocity relative to the surface, it follows that if A and B are the inlet and outlet respectively, the friction at these two places is in the ratio v_2^2 to v_1^2. Knowing f, the mean friction, these can be determined approximately. Let them be f_1 and f_2. If $A\,B$ (Fig. 86) is the profile of a blade erect perpendiculars f_1 and f_2 at A and B. Join these by a gradual curve $a\,c\,b$ of steadily-increasing perpendicular distance from A to B. Then this is a curve of friction; and by projecting the perpendiculars to this curve on to the plane of rotation we get the axial

components of these frictional forces. In this way we obtain the curve $m\ n\ r$. The mean height of this curve from the blade profile $A\ C\ B$ gives the mean axial friction. If this is $0 \cdot 6$ of the mean friction (f), and if we have also

$$\alpha = 65^\circ$$
$$\beta = 45^\circ$$
$$V = 250\text{ft. per second}$$
$$2k = 0 \cdot 322$$

Then

$$v_1 = 663$$
$$v_2 = 516$$

and the friction end thrust is

$$\frac{\cdot 6 \times \cdot 322\ (663 - 516)}{32 \cdot 2} = 0 \cdot 88.$$

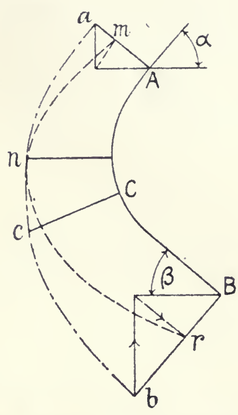


FIG. 86.—FRICTION ON TURBINE BLADES.

The method of balancing each section of the turbine by means of its own balance piston secures a good balance at light loads. Where only a single-balance piston is used its high-pressure side should be connected to an intermediate section of the turbine.

Most Efficient Blade Angles for a Many-stage Turbine.—The usual method of determining the most efficient blade angles is by taking the kinetic energy in the exhaust from each set of moving blades to be entirely lost, and, moreover, to be the only loss. This method is entirely wrong.

In the first place, the kinetic energy in the exhaust from a set of blades is seldom entirely lost, and should, indeed, be made use of in the next set of blades, so that it is only the final exhaust from the last row of moving blades which matters, and this should not exceed from 3 to 6 per cent. of the total available work in the steam as it comes from the boiler. The most serious losses are those due to friction, and leakage past the blades, or (in an impulse turbine) the diaphragms; this latter being of less importance than the former, save in very small turbines.

The former (friction) loss we reduce by choosing suitable blade angles and peripheral speeds; the latter (leakage) by choosing suitable diameters and peripheral speeds for the rotor. To a certain extent these losses are antagonistic, a small leakage loss being usually accompanied by a large friction loss.

Let v_o = mean velocity relative to blade surfaces.
h = blade height (average in a given section).
n = number of blades per foot of periphery.
m = width of blades (axial).
d = diameter of blade circle.
k = a constant, varying between 0·9 and 0·96.
v = axial velocity.

Then the frictional loss in each stage is proportional to the product of the square of the velocity, the perimeter of the passages and the length of the passages, and this is equal to

$$K\, v_o^2 \times \text{perimeter} \times v_o \times \text{time in stage.}$$
$$= K\, v_o^3 \times \text{perimeter} \times \text{time.}$$

Now the time in one stage

$$= \frac{\text{length of steam path across the stage}}{\text{mean velocity in stage}}$$

$$\text{Perimeter} = 2\,\pi\, d\, k + 2\, h\, n\, \pi\, d$$
$$= 2\,\pi\, d\,(k + h\, n)$$

But n is approximately inversely proportional to m and to h, so that the mean perimeter of the passages is a constant and can be left out of our calculations. Hence we may say that the friction loss per stage is proportional to

$$v_o^3 \times \text{time in one stage.}$$

And the *percentage* friction loss is proportional to

$$\frac{v_o^3 \times \text{time in one stage}}{\text{work done in one stage.}}$$

Case 1.—Reaction Turbine with Fixed and Moving Blades of the same Shape.

Time in one stage $= \frac{c_1 m}{v} = \frac{c_2 h}{v} = \frac{c_3}{v^2}$

approximately ; c_1, c_2, and c_3 being constants.

Work done in one stage $= \frac{v_1^2 - v_2^2}{g}$

Hence friction loss per cent. is proportional to

$$f = \frac{(v_1 + v_2)^3}{(v_1^2 - v_2^2)\, v^2}$$

From this formula the curves in Fig. 87 have been calculated. The curves marked $\beta = 20$, $\beta = 30$, &c., show the friction losses for different inlet angles α when

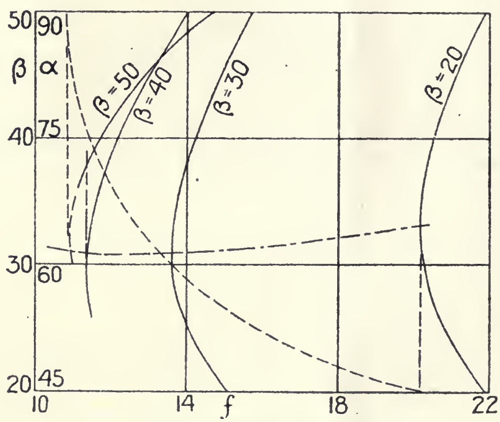


FIG. 87.—REACTION TURBINE.

the outlet angle β has the values marked on the curves. The approximately horizontal dotted-line curve passes through these first curves at the values of α which give the lowest losses. The other dotted curve, which is approximately hyperbolical in shape, gives the relationship between the values of β and f (friction) for these best

values of α. When this dotted curve becomes approximately vertical then we cannot further greatly decrease the friction loss and hence the curve gives us the best value of β at this point. Referring to our first (approximately horizontal) dotted curve, we obtain the corresponding value for the inlet angle α. These angles, then, are the most efficient blade angles so far as friction loss is concerned. In this case these angles have the values :—

$$\alpha = 60 \text{ to } 65^\circ$$
$$\beta = 40 \text{ to } 45^\circ$$

These blade angles necessitate rather high steam velocities and accurate formation of the blade shapes. Owing to the high velocities the blade radial heights will be rather small, which means that the clearance leakage becomes relatively more important, so that in small turbines it will be advisable to adopt somewhat lower values for the outlet angle β, the corresponding value of α being obtained from the curves. It is common practice with makers of reaction turbines of this type to make the inlet angle α equal to 90°. This, however, whilst enabling low velocities to be used for the steam and thus reducing the clearance leakage (by lengthening the blades) is not the most efficient, and necessitates a long rotor.

In order to reduce the final loss of kinetic energy to the exhaust the last row or two sometimes have a considerably smaller outlet angle (say 20°) than the other rows of blades.

(2) **Impulse Turbine.**—This case differs from the last in that the velocities relative to the surfaces are not the same in the fixed and moving blades. We shall assume that the relative velocity in the moving blades is the same at inlet and outlet, and hence the average velocity will be approximately proportional to

$$v_0 = v_3 + v_1 + 2\, v_2.$$

Now, work done per stage $= \dfrac{V}{g}\,(2\, v_2 \cos \alpha)$, so that the percentage friction loss is proportional to

$$f = \frac{(v_1 + 2\, v_2 + v_3)^3}{v^2\, v_2 \cos \alpha},$$

Fig. 88 illustrates the results obtained when β and a are varied. From the figure it appears that the best values for β and a are approximately 35° and 48°. In order to secure the same velocity at outlet and inlet we must have a slight pressure drop and volumetric increase for the steam in the moving blades. If there is no such pressure drop the velocity at outlet will be $k\ v_2$ where k is a constant, probably about 0·9 or even higher, and we have approximately

$$f = \frac{(v_1 + 1 \cdot 9\ v_2 + v_3)^3}{v^2\ v_2 \cos\ a}$$

The value of v_3 is not now the same as before, but is less. Fig. 89 shows the results for this case. The best angles a, β, and ϕ (inlet to nozzles) are about 38°, 51°,

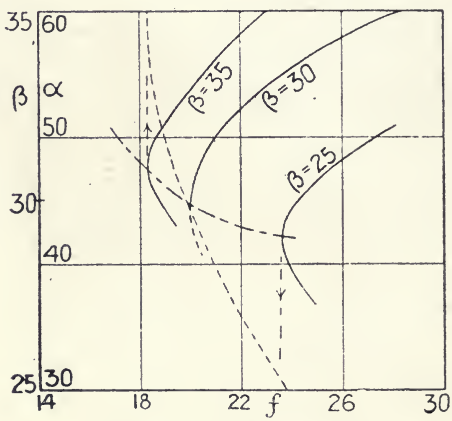


FIG. 88.—IMPULSE TURBINE.

and 90° respectively. The nozzle inlet angle in the former case should be 87°. In these investigations we have neglected any possible loss of kinetic energy in the outgoing steam. The occurrence of such loss would make somewhat smaller angles than the above desirable. According to Prof. Rateau, the angles a, β, and ϕ for the Rateau turbine are 30°, 20°, and 95°, These seem distinctly too small (except the nozzle inlet angle) (see Fig. 19).

In the previous examples we have assumed that the width of the blade is proportional to the height. This is

never quite attained in practice, although it is not so very different from the proportions obtaining in some turbines

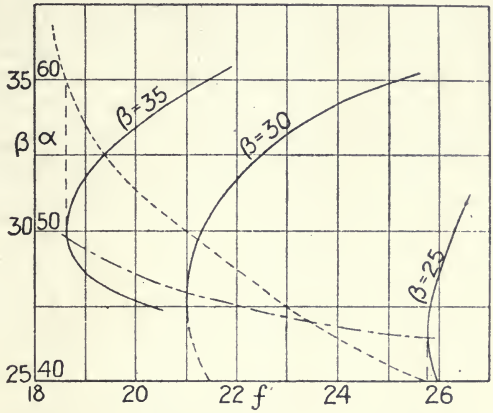


FIG. 89.—IMPULSE TURBINE: REDUCED OUTLET VELOCITY.

of the Parsons type. In impulse turbines the axial width of the blades is seldom varied in this manner, and

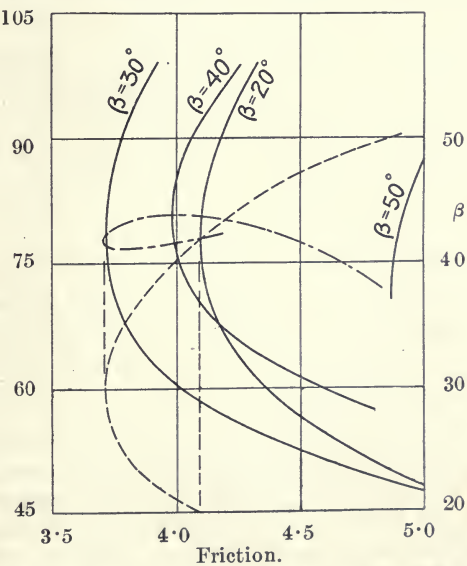


FIG. 90.—REACTION TURBINE; CONSTANT BLADE WIDTH.

is, indeed, usually constant in any one cylinder.

These facts materially affect the best values for the

blades. In Fig. 90 we have curves similar, in a general way, to those of the three preceding figures, illustrating the best angles for reaction turbines when the blade width is constant. It will be seen that the best angles in this case are about 75° or 80° and 30°. How far this case can be taken as representative of actual practice is somewhat problematical. It would seem, however, that these values approximate to the conditions in small turbines, whilst in large turbines the variation of the

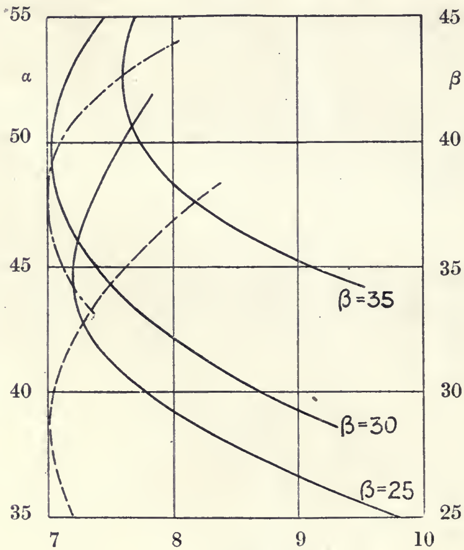


FIG. 91.—IMPULSE TURBINE; CONSTANT BLADE WIDTH.

blade width with its height, whilst not by any means strictly proportional, is nevertheless considerable.

Perhaps angles of about 70° and 35° would be suitable for fairly large turbines.

In Fig. 91 we have curves showing the losses in an impulse turbine with constant blade width and no loss of velocity in the moving blades. The best angles seem to be about 27° to 30° for the outlet from the nozzle and about 47° for the moving blades. In Fig. 92 we have the curves for an impulse turbine with loss of velocity—5 per cent.—and constant blade width. The best angles in this case are about 30° and 50°.

In reaction turbines of small size, the necessity of having a fair blade length is an argument in favour of the smaller angles. In impulse turbines the loss at the exhaust from each wheel is an argument against the use of large angles. For a medium-sized turbine of this type angles of about 25° and 40° would probably be most suitable.

A close comparison of the relative efficiencies of the impulse and reaction types when designed to give the best results is impossible; there is not very much to choose, however, and the difficulty of making accurate comparisons renders the results not at all certain. The

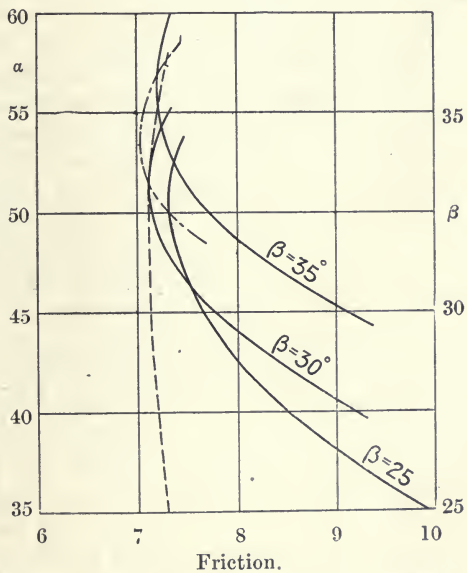


FIG. 92.—IMPULSE TURBINE; REDUCED OUTLET VELOCITY; CONSTANT BLADE WIDTH.

reaction machine seems to suffer less from having its blade angles improperly chosen than does the impulse machine. Then, again, the losses at the exhaust of each stage are probably less for the reaction machine, its leakage losses will be considerably greater, but there will be no "disc" friction due to the rotation of the turbine discs in the steam chambers.

(3) **Curtis Turbine.**—As we have just shown, with a given wheel speed there are certain blade angles and steam

velocities which give the best results. In a Curtis turbine the wheel speed remains the same for all the moving blades in any one stage, but its steam velocities and blade angles must vary so that only one out of the several sets can have the most efficient blade angles. Consequently, the losses in a Curtis turbine will, with the exception of disc friction, be greater than in an impulse turbine of the Rateau type. By using somewhat higher wheel speeds the Curtis turbine reduces this discrepancy. That these results are borne out in practice is evidenced by the fact that whereas formerly four and even more wheels were used in each stage, now only two or occasionally three are employed.

Mr. H. F. Schmidt† has attempted to analyse the losses in a Curtis, and also in a Westinghouse-Parsons turbine. These analyses are not to be accepted without reservation, but they are certainly interesting. From tests on a 500 kw. Curtis turbine he obtained the following results: Work available at shaft, 56 per cent.; loss in final velocity of steam, 14 per cent.; friction of wheels in chambers, 6·2 per cent.; nozzle losses, 6 per cent.; losses in buckets, spreading, and leakage, 14·8 per cent.; radiation, 3 per cent.; total, 100 per cent.

For a 1,250 kw. turbine on the lighting service of the New York subway, the figures obtained were: Energy available at turbine shaft, 62·8 per cent.; friction of drum and buckets, 6·3 per cent; radiation, 7 per cent.; loss by final velocity of steam (assumed), 12 per cent.; losses in buckets and guides, friction, and leakage, 11·9 per cent.; total, 100 per cent. The loss by final velocity of the steam was admittedly doubtful and in the present writer's opinion ought not to be larger than about 5 per cent. at the outside. If we take it at 4 per cent., then the losses in buckets and guides, friction and leakage, run up to 18·9 per cent.

The superior efficiency of the Parsons turbine over the Curtis turbine is to be partly accounted for by the discrepancy in size. In any case, too much weight should not be given to the results of a comparison of one turbine of each type only.

† "Street Railway Journal," June 25th, 1904.

Best Peripheral Velocity.—With fixed blade angles the steam velocities will be proportional to the peripheral speed so that the percentage friction loss is proportional to

$$\frac{V^3 \times \text{perimeter of passages}}{V^4}$$

$$= \frac{\text{perimeter}}{V} = \frac{\text{surface}}{m\ V}$$

First let us take the case where the diameter is fixed, the velocity V being varied by altering the revolutions. The perimeter will vary but little with a variation of speed, for although the blade height will be reduced with an increase in speed, yet the number of blades will correspondingly increase. Hence the *friction loss is roughly inversely proportional to the peripheral speed.* As an offset to this gain we have in the reaction turbine an increased loss by leakage over the ends of the blades due to the proportionally greater radial clearance at high speeds. This is the reason why the high-pressure end of a turbine, where the blade lengths are naturally small, is generally of less diameter (thus giving a less peripheral speed) than the low-pressure end. In an impulse turbine the disc friction loss runs up as the cube of the speed, and thus reduces the gain from the use of high peripheral velocities. Also the exhaust loss is greater at high speeds. Next suppose that the revolutions per minute are fixed, the diameter being varied to vary the peripheral velocity. The loss is proportional to the surface areas, that is to

$$(2\,\pi\,d\,k + 2\,h\,n\,\pi\,d)\,m.$$

Now

$$V = k_1\,d$$

$$h = \frac{k_2}{d\ V} = \frac{k_3}{V^2}$$

where k is a constant.

Approximately m is proportional to h, and inversely proportional to n, so that the loss is proportional to

$$\frac{1}{V}$$

and hence a large diameter is desirable. The limitations imposed upon an increase in diameter have been discussed.

Considerations of strength also limit the peripheral velocity in large turbines where the leakage is of less importance.

The following table (Table VII.) gives some of the rotational speeds usually employed.

TABLE VII.

Kw.	50	100	500	1,000	2,000	5,000
C. A. Parsons ...	5,000	3,500	3,000	1,800	1,000	750
Westinghouse ...	—	—	3,600	1,800	—	750
Curtis	—	—	1,800	—	750	500
Rateau	—	—	2,200	1,500	—	—
Brown-Boveri ...	—	—	—	—	1,500	—
Zoelly	4,000	4,000	3,000	1,500	1,500	1,000

The Brown-Boveri turbine is of the Parsons type. A 3,000 kw. turbine by Brown-Boveri & Co., at Frankfort, runs at 1,360 revs. per minute. A table showing the speeds of De Laval turbines was given in Chapter II. The actual speed used in any individual case depends to some extent on the frequency of the dynamo, if an alternator.

The peripheral velocities in a Parsons turbine usually vary from about 100ft. or 150ft. per second at the high-pressure end, up to 300ft. or 350ft. per second at the low-pressure end. The same velocities are used in nearly all makes of the Parsons turbine.

The peripheral velocity of a Curtis turbine is usually 420ft. per second. Rateau employs a peripheral speed between 300ft. and 400ft. per second. Zoelly goes considerably higher, and in the Riedler-Stumpf and De Laval turbines velocities exceeding 1,000ft. per second are employed. The peripheral speeds of marine turbines are usually much lower than the above, on account of the inefficiency of very small, high-speed propellers, and the danger of the propeller racing.

Table VIII. gives the speeds usually adopted in marine work for the blades. Table IX. gives some blade speeds adopted in electrical work, these data being taken from a recent paper by E. M. Speakman.*

TABLE VIII.

Type of Vessel.	Peripheral Speed of Blades.	
	High Pressure.	Low Pressure.
Fast mail steamers	70 — 80	110 — 130
Medium mail steamers	80 — 90	110 — 135
Channel steamers	90 — 105	120 — 150
Battle-ships and large cruisers...	85 — 100	115 — 135
Small cruisers	105 — 120	130 — 160
Torpedo craft	110 — 130	160 — 210

TABLE IX.

Rated Output in Kilowatts.	Revolutions per Minute.	Peripheral Speed of Blades.	
		H.P. Section.	L.P. Section.
5,000	750	135	330
3,500	1,200	138	280
2,500	1,360	125	300
1,500	1,500	125	360
1,000	1,800	125	250
750	2,000	125	260
500	3,000	120	285
250	3,000	100	210
75	4,000	100	200

Blade Sections.—The cross-section of the blade must be sufficient for purposes of strength and so shaped as to give a correctly-formed steam passage.

(1) **Impulse Turbine.**—For the sake of simplicity we shall assume a symmetrical blade. We shall make the width of the passage between two consecutive blades constant, when measured at right angles to the direction

* "Dimensions of Steam Turbines for Marine Work." Paper read before the Institution of Engineers and Shipbuilders of Scotland, October 24th, 1905.

of flow. Any changes in the cross-sectional area of the steam jet are allowed for by suitably altering the radial depth of the passages. Referring to Fig. 93, *N M R* is the axis of symmetry, *A* and *B* the edges of two consecutive blades. The short line *C D* is the direction

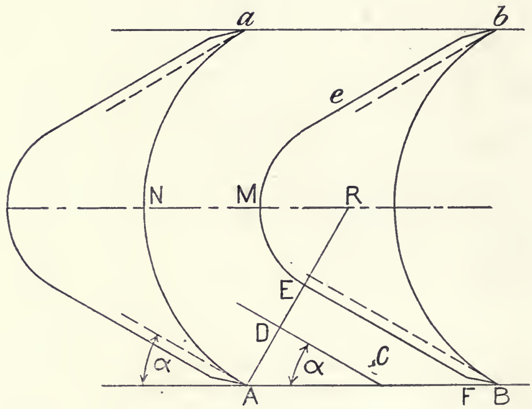


FIG. 93.—IMPULSE BLADE DESIGN.

of flow of the steam at outlet. Draw *A R* perpendicular to *C D* and cutting the centre line in *R*. With *R* as centre and radius *R A*, describe the arc *A N a*. This is the concave profile of the blade. Draw *E F* parallel to *C D* and a little nearer *C D* than a parallel through *B*. If *F E* cuts *R A* in *E* describe the arc *E M e* with *R* as centre. Then *M E F B* is half the convex profile of the blade, the other half being similar. By this construction the perpendicular width of the steam passage is kept constant. The portion *F B* of the back face is merely a small shamfer intended to reduce the eddy losses at inlet and outlet. If we had made *E F B* a straight line it would not then be parallel to *C D*, for the direction of the steam is determined by the *average* angle of the blades at outlet and not by the back or front face alone. It is common practice to make *E F B* straight, although it does not give quite such a good blade section either as regards strength or "hydraulic" properties as the one described above. In any case, it is desirable to make the inlet edge of the blade blunter than the outlet edge, as the former has to work under severer conditions than the

latter, and indeed practically the whole of the erosion of the blades takes place at the inlet edge. The relation of the pitch of the blades ($A\ B$) to the axial width depends on the angles employed and is limited by three conditions.*

If the pitch or distance between consecutive blades is large the curvature of the face $E\ M\ e$ will be sharp, and this will cause losses in the steam. If, on the other hand, the pitch is small, the curvature $E\ M\ e$ now being much gentler, the surface of the blades and consequently the friction loss will be considerably increased. Further, the blade may now become too thin and weak. The cause of the losses referred to when the curvature is sharp is the alternate increase and decrease in the steam velocity which such sharp curves cause. As the steam passes round the curve the velocity on the inside of the curve will be less than on the outside of the curve, so that there will be a fall in this (inner) steam velocity, and after passing the curve there will be an increase in the velocity. The action is analogous to the passage of water through a bent pipe.

The pitch or circumferential distance between consecutive blades usually varies from 0·4 to 0·6 or 0·7 of the axial width of the blade. Where, as in the Rateau turbine, sheet-metal blades are employed this ratio can be exceeded and the surface friction reduced. The argument usually employed against the sheet-metal blade may be stated as follows. Steam turns a corner (or rather a curve) in much the same way that a body of well-drilled soldiers does, a line drawn perpendicularly across the path of the steam at one point being carried along with the steam, so that it always remains perpendicular to the path. Hence at a curve the steam on the inner side is going at a less velocity than that on the outside, thus giving rise to some loss. With sheet-metal blades the width of the path obviously varies, since we have now a constant circumferential width, which will give rise to an expansion followed by a compression as the steam turns the curve, a source of considerable loss, greater probably than where a constant width of path is

* More properly this should be called the circumferential pitch, as the term pitch is usually used to denote the axial distance between the centres of two consecutive sets of moving blades.

maintained. It is doubtful, however, whether the steam does move in this geometrical fashion. If it does not then except where considerations of strength demand a thick blade the sheet-metal blade with shamfered edges is probably the better of the two. It ought not to be difficult to settle the matter experimentally.

From the geometry of the blade profile we see that if the curvature of the back face has a constant ratio to the curvature of the front face, the ratio of the axial width to the circumferential pitch is directly proportional to the sine of twice the blade angle. If the radius of the concave face is twice the radius of the convex face the ratio of axial width to circumferential pitch will be

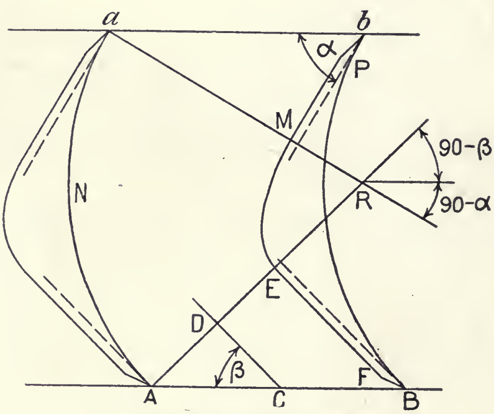


FIG. 94.—REACTION BLADE DESIGN.

0·783 with angles of 20° ; 0·578 with angles of 30° ; and 0·507 with angles of 40°. The values are probably as large as is desirable in practice.

A somewhat similar relation between this ratio and the angles holds for a reaction turbine.

(2) **Reaction Turbine** (Fig. 94).—The general method of design is the same as that for an " impulse " blade, the geometry being a little different. As before, A and B are the outlet edges of two blades, $C\ D$ is the direction of the steam as it leaves the blades. Choose a centre R, draw $R\ a$ perpendicular to the inlet edge, and describe the arc $A\ N\ a$ with centre R. $A\ N\ a$ is the concave profile of the blade. As before, draw $E\ F$ parallel to $C\ D$

and meeting $A R$ in E. Draw $M P$ parallel to the direction of the inlet steam to meet $R a$ in M. Join $M E$ by a curve, giving a gradual change in the width of the steam path. Then $b P M E B$ is the convex profile of the blade. As before, the curvature of the face $M E$ should not be too sharp. Instead of the shamfer $P b$ we may have $b P M$ straight. In this case the true inlet angle at b is obtained by drawing a line bisecting the angle made by the two faces at b, and similarly for the outlet angle. The design of the nozzles in a many-stage impulse turbine follows the same lines. The outlet cross-section must be less than the inlet cross-section, provided that the pressure at the outlet is roughly more than half that at inlet. With greater pressure drops the nozzles must be designed in the manner previously described, but should be placed close together in two or more symmetrical groups, so as to reduce the boundary losses.

It is usual to make the front and back profiles of the blades of reaction turbines complete curves from inlet to outlet. There are thus no straight lines in these profiles. The inlet edge is made somewhat blunter than the outlet edge. The usual outlet angle is between 20° and 30°, although with long blades this is frequently increased to about 40°.

Strength of Blades.—The blade is subjected to stresses due to centrifugal force and the steam reaction, both being greatest at the root of the blade ; hence so far as strength is concerned the blade may be made thinner at the outer end than at the root. This makes an expensive blade, and is seldom if ever of any considerable value ; indeed, some designers consider it bad to have a wider steam path at the outer end of the blade than at the inner. The reaction on the blade is easily calculable from the speed and power of the turbine. First we obtain the turning moment on one set of moving blades, and dividing by the number of active blades in the set we obtain the pressure in the direction of rotation on one blade. The end thrust on the set of moving blades, and hence the end thrust on each blade, can be calculated as was explained earlier on in this chapter. Combining these two thrusts we obtain the resultant thrust in magnitude

and direction. Consider this as concentrated at the middle of the blade, and we can then calculate the maximum tension (at the root of the blade) due to the bending moment thus set up. Add to this the centrifugal force of the whole blade divided by the bottom cross-section and we arrive at the greatest stress in the blade, which is a tension.

For instance, in a 2,000 kw. reaction turbine, the steam used per hour is 36,000lbs., or 10lbs. per second. Consider one of the last set of blades at the exhaust end; the "indicated" work for the set is, in foot-pounds per second:—

$$\frac{10 \times V}{g} (v_1 \cos \beta + v_2 \cos \alpha)$$

$$= \frac{10 \times 360}{32 \cdot 2} (675 + 314) = 110{,}500$$

Dividing by V we obtain the tangential pressure, which is 307lbs. in the plane of rotation. Combining this with an end thrust of, say, $10 \times 12 \cdot 32 = 123$lbs. we obtain a resultant pressure on the blades of 331lbs., making an angle of 24° to the plane of rotation. Suppose, further, that the pressure at the exhaust is 1lb. absolute, and that the dryness is 80 per cent. The volume of steam passing per second will be $330 \times 0 \cdot 8 \times 10 = 2{,}640$ cub. ft., and the axial velocity is 360ft. per second, so that the cross-sectional area of the passages is 7·33 sq. ft. If the turbine makes 1,000 revs. per minute the diameter must be 6·88ft., and the radial depth of the vanes 4·08in. If there are 331 blades in the set, all of them acted upon by the steam, the pressure on each blade will be 1lb., and the maximum bending moment on the blade will be $1 \times 2 \cdot 04$in.-lbs. We can perform a similar calculation for each row of moving blades. In an impulse turbine with partial peripheral admission it must be remembered that only a portion of the blades receive the turning effort of the steam.

The centrifugal stress in pounds per square inch for a blade of length l inches, any cross section, weighing w pounds per cubic inch, having a peripheral speed V, and describing a circle of diameter d feet, is

$$0 \cdot 0622 \frac{w\,l}{d} V^2$$

if $w = 0{\cdot}305$ (brass or delta metal) then the value of the stress is

$$0{\cdot}019\frac{l}{d}V^2$$

The stress due to centrifugal force on the blades used in the last example will therefore be 1,460lbs. per square inch.

In order to calculate the stress due to bending moment we must determine the moment of inertia of the blade section. Methods of doing this are explained in books on mechanics. We may, however, just indicate the method. In Fig. 95 $X\ Y$ is the line of action of the

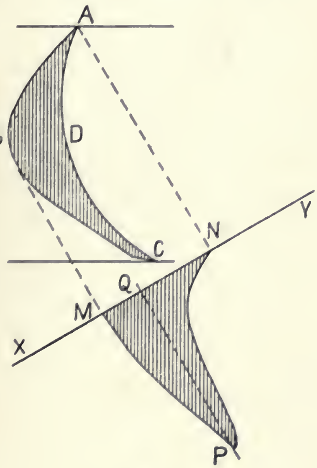


FIG. 95.

resultant pressure on the blade, and $A\ B\ C\ D$ is the blade cross-section. Project this area on to the axis $X\ Y$ so that the figure $N\ M\ P$ has the same width at all points as the original cross-section. This does not in any way alter the area, but makes the figure simpler to deal with. Determine the centre of gravity of the area $M\ N\ P$, and draw the neutral axis $P\ Q$ through the centre of gravity and at right angles to $X\ Y$. Divide the figure $M\ N\ P$ into strips of areas a_1, a_2, &c., by lines parallel to $P\ Q$, and let the distances of the centres of gravity

of these areas from $P\,Q$ be x_1, x_2, &c. Then the moment of inertia of the blade section is

$$I = a_1\,x_1^2 + a_2\,x_2^2 + \ldots\ldots\ldots\ldots$$

all measurements to be in inches.

Then if f is the maximum tension in the blade due to the bending moment M which we have previously determined, we know that

$$f = \frac{M \times QN}{I} \text{ lbs. per square inch.}$$

Add to this the stress due to centrifugal force and we obtain the maximum tension in the blade. As a rule, the blade will be amply strong enough, and we decide its cross-section on mechanical grounds. The section must be properly shaped, and of sufficient size to enable all its curves and angles to be quite definite, and above all the blade must make a sound mechanical job when fitted into position. The length of a blade is seldom more than 10 times the width and not often less than three times or twice the width. Very long blades are usually supported (in a Parsons turbine) by means of a wire threaded through them close to the outer end. Messrs. Willans & Robinson fit a channel section over the heads of the blades, the main object being, however, to reduce the "clearance" leakage past the blade ends. (Figs. 55 and 104.)

Arrangement of Blades.—It is highly important that the arrangement of the blades be given careful consideration. Where partial peripheral admission is employed it is of the greatest importance to have the fixed blades or nozzles in consecutive stages, in such relative positions that the steam from the moving blades enters easily into the fixed blades as is illustrated in Fig. 96. For instance, if the axial velocity of the steam is equal to the peripheral speed of the moving blades the steam, whilst crossing between two adjacent sets of fixed blades, will have moved round the circumference of the blades by a distance equal to the distance apart (parallel to the shaft) of the adjacent faces of the two sets of fixed blades. Consequently the centre of the fixed-blade openings in the latter set must be in advance of the centre of the

previous set of fixed blades by an amount equal to the distance across the space between them. If this is not attended to the steam leaving the moving blades will run up against a solid portion of the fixed ring or dia-

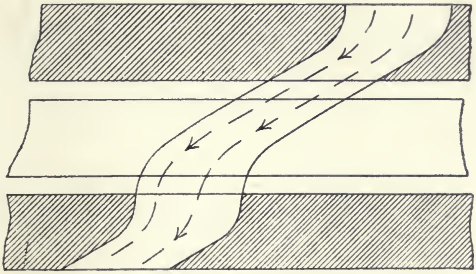


FIG. 96.—SHOWING PROPER RELATIVE POSITIONS OF FIXED BLADES IN CONSECUTIVE STAGES.

phragm, and the bulk of its kinetic energy will be lost on account of the eddies set up in the steam. This is illustrated in Fig. 97. Where, as is frequently necessary, the length of the peripheral openings increases from one diaphragm to another there is bound to be some loss on this account, but it is reduced to a minimum if the spacing

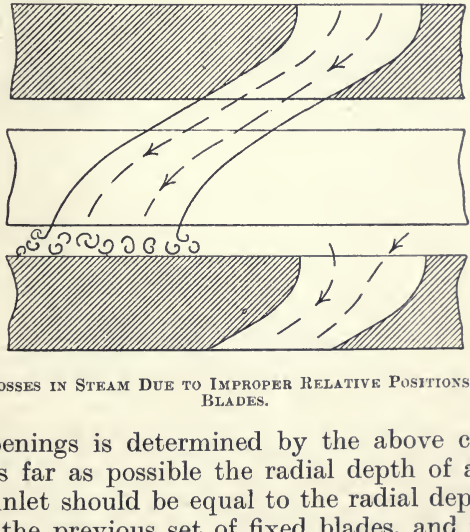


FIG. 97.—LOSSES IN STEAM DUE TO IMPROPER RELATIVE POSITIONS OF FIXED BLADES.

of the openings is determined by the above considerations. As far as possible the radial depth of a moving blade at inlet should be equal to the radial depth of the outlet of the previous set of fixed blades, and its radial

depth at outlet should be equal to the radial depth of the inlet to the next set of fixed blades. If, for instance, the moving blade extends radially inwards beyond the fixed blades then the jet of steam issuing from the fixed blades

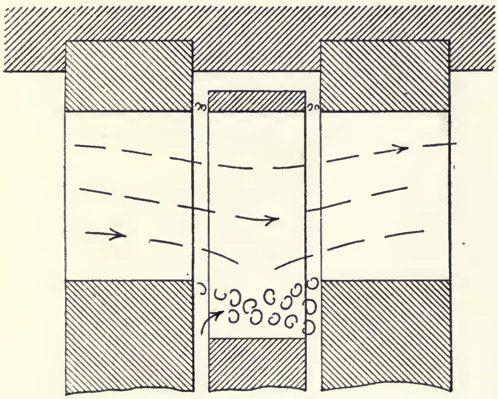


Fig. 98.—Eddy Loss Due to Improper Blade Depth.

will become entangled with the comparatively stagnant stream in which the moving blades revolve, and thus give rise to considerable eddy losses as is illustrated in Fig. 98. A somewhat similar action occurs when the

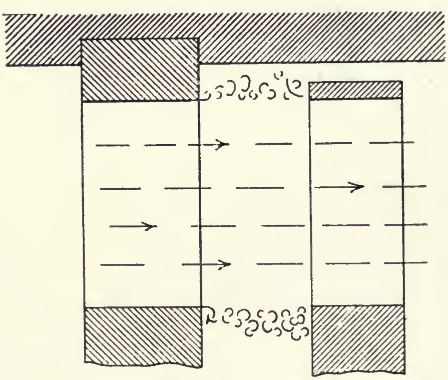


Fig. 99.—Eddy Loss Due to Large Axial Clearances in Impulse Turbine.

axial clearance between the fixed and moving blades of an impulse turbine is large, as is illustrated in Fig. 99. This action is not nearly so marked in a reaction turbine, because in this case there should be practically no stagnant

steam, all the steam forming part of one or more streams or belts flowing between the stop valve and the exhaust. This will be evident from a consideration of Fig. 101, which shows how the blades of a reaction turbine should be attached to the rotor and casing. Fig. 100 shows how they should *not* be attached; that is to say, the blade openings should extend right down to and be flush with the casing or the rotor as the case may be. In some of the more recent Zoelly turbines the moving blades are considerably longer than the fixed blades, a

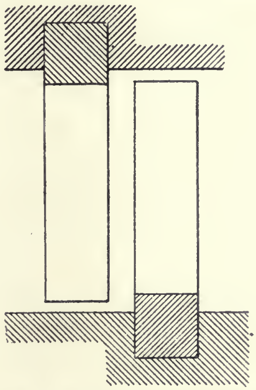


Fig. 100.—Incorrect Attachment of Blades of Reaction Turbine.

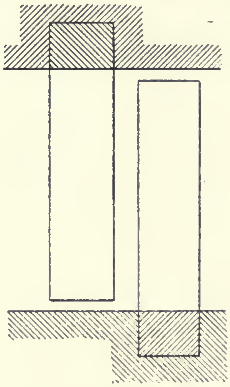


Fig. 101.—Correct Attachment of Blades of Reaction Turbine.

slight percentage of efficiency being sacrificed in order to simplify the construction of the machine, and in particular to reduce the linear speed of the rim of the discs to which the blades are attached.

It may be thought that since impulse turbines usually have partial peripheral admission at the high-pressure end it would be advisable to adopt the same plan in reaction turbines. There is, however, a difference between the two cases. In the impulse turbine there is no difference in pressure between the two sides of a wheel, so that except where it is forcibly ejected from a nozzle the steam has no tendency to flow through the blades from one side of the wheel to the other. In a reaction turbine there is a difference in the pressures at the two sides of the wheel (a wheel here meaning a set of moving blades), so that at *all* parts of the wheel, and not

merely between the openings in consecutive fixed diaphragms, there will be a flow of steam through the blades from one side of the wheel to the other, and except for that portion of the steam in the line of flow between the diaphragm openings the steam will discharge itself against a blank face of the diaphragm, and its kinetic energy will be churned into heat energy. Further, in order to enable this flow to take place all round the wheel circumference, the steam leaving a diaphragm opening will be diverted in a circumferential direction, and much of its kinetic energy thereby lost. This can be partially prevented by making the axial clearances between the fixed and moving blades small. The moving blades

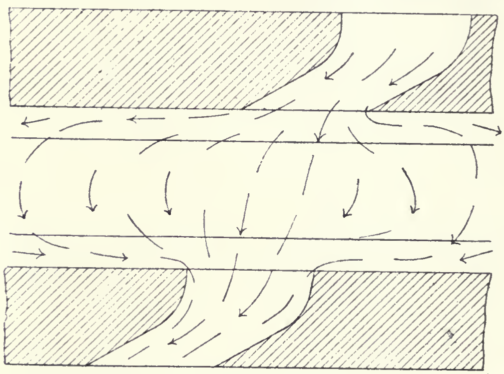


FIG. 102.—SPREADING DUE TO PARTIAL PERIPHERAL ADMISSION IN A REACTION TURBINE.

in the case of a reaction turbine with partial peripheral admission virtually form a sudden enlargement of cross-section in the passage area of an expanding nozzle. This sudden enlargement is accompanied by considerable losses, which are aggravated in this case by the obstruction offered by the moving blades. Some idea of the actions taking place in such a turbine is given by Fig. 102. If the small size of the turbine makes it necessary to adopt partial peripheral admission (for a reaction turbine) it is imperative that the axial clearances should be made as small as possible. Where there is, however, full peripheral admission it is a common rule to make the axial clearance equal to about one-half of the axial width of the blades. This will, of course, be different at different parts of the turbine.

The axial pitch—commonly simply called pitch—or distance between the centres of consecutive sets of moving blades is then usually equal to three times the axial width of the blade. The radial clearance is dependent on a number of conditions which will be more fully discussed later. It is very seldom less than 0·02 of an inch and is usually about 0·012 of an inch per foot of rotor diameter.

It is probable that the efficiency of turbines could be somewhat increased by greater attention to the minutiæ of the blade constructions; as, for instance, by making the radial depths of the blades to correspond at all points to those determined by calculation and in particular to allow for the expansion in the individual blades. How far these improvements would justify the extra cost of manufacture is certainly very problematical, and can only be settled by experimenting. One objection to the use of soft caulking pieces between blades, as adopted in most reaction turbines, is the difficulty of making the caulking piece (after caulking) flush with the surrounding metal without going to undue trouble. This ought not, however, to be really serious to overcome, and turns mainly on cutting the right size of caulking pieces. A further objection to the use of brass caulking pieces laid in a groove in a steel rotor is that the brass expands more than the steel for a given rise of temperature (about 50 per cent. more), and thus tends to force the blades out of the rotor. For this reason some makers use steel caulking or distance pieces. With such an arrangement the extra expansion of the brass blades is sufficient to tightly compress the blades and caulking pieces together without, it is claimed, lifting out of the groove or permanent distortion.

Material for Blades.—The qualities usually desirable in a material to be used for turbine blades are :—

(1) Capable of being rolled or drawn into strips of the same section as the finished blade.

(2) Hard enough to resist abrasion by the high-velocity steam.

(3) Fair strength.

(4) Must not rust or corrode under working conditions.

(5) Cheapness.

Of course, the conditions under which the blades will have to work are very different in different types of turbines, so that a material which is quite suitable for a reaction turbine may not be suitable for a single-stage impulse turbine. Then, too, the effect of superheat has to be considered. In general it is advisable to assume that at some time or other superheated steam will be used in the turbine, although an exception to this rule can be made in the case of a marine turbine, there being very little probability that a superheater will ever be added to a vessel after she has once received her boilers. When superheated steam is supplied the superheat is all lost long before the exhaust is reached, so that only the blades at the high-pressure end need be designed for superheated steam.

Gun-metal, brass, and copper are unsuitable for high superheat, and are generally replaced by a nickel alloy. The bulk of the blades of reaction turbines are of brass, containing about 3 per cent. of tin. Those for high superheat usually contain about 80 per cent. of copper and 20 per cent. of nickel.

The blades of a De Laval turbine are of hard steel, whilst those of the Riedler-Stumpf are milled out of the solid rim of a nickel-steel disc. The blades of a Zoelly turbine are of nickel steel accurately machined and polished, whilst those of the Curtis turbine are cut from a solid ring of steel. All the above turbines use high-velocity steam. Delta-metal, phosphor-bronze, and brass are very commonly used for blades in many-stage turbines where the velocities are low.

There seems to be no doubt but that there is an appreciable amount of wear on the inlet edges of the blades in all steam turbines. This wear does not, however, seem to have a very marked effect on the steam consumption provided it is not excessive. It is desirable, in the absence of definite knowledge that the effect of this wear will always be small, that the blades should be so fixed that they can easily be taken out and replaced.

As wet steam is such an excellent thing for wearing the blades it is desirable where saturated steam is used that a steam separator should be used in front of the

admission valve. Superheated steam cannot of course be wet, and hence one of its advantages.

As regards the strength of the material. The centrifugal stress alone frequently assumes considerable magnitude, and may be as much as two tons per square inch at the low-pressure end of a Parsons turbine, although such a value is unusual. In the De Laval turbine very high centrifugal stresses are attained, as high as 15 tons per square inch in some cases although usually less, being in fact about 10 tons per square inch in a 300 b.h.p. turbine and somewhat less in the smaller sizes. The stress due to centrifugal force increases from the outer end to the root of the blade, and hence may be reduced by tapering the blade so that the thickness is greatest at the root. The stresses due to bending moment are greater (in general) the higher the blade velocity, and are somewhat greater in impulse than in reaction turbines.

The stresses on the blades of a De Laval turbine—and also those of other impulse turbines—are fluctuating in character. The centrifugal stress is constant, but the bending stresses pass from zero to a maximum and back to zero every time the blade passes an active steam nozzle or group of nozzles. Thus in a 300 h.p. De Laval turbine making 10,600 revs. per minute and having eight nozzles open, the bending stresses would have a frequency of 1,413 per second or 84,800 per minute. Now it is well known* that these repeated stresses are much more dangerous at high than at low frequencies, so that it is not difficult to understand why the blades of these turbines frequently break off after some time in service.

Although the centrifugal stress does not vary, yet it makes the variable bending stresses much more dangerous than they otherwise would be, for it has been proved experimentally, and is confirmed by the theory of fatigue expounded in the paper referred to, that a variation of stress between, say, 10 and 15 tons per square inch is very much more dangerous than a variation between, say, 0 and 5 tons per square inch, although the actual range of the variation of stress is the same in the two cases. This is

* "On Phenomena Due to Repetitions of Stress, and on a New Testing Machine." By Frank Foster, B.Sc. Proc. of the Manchester Literary and Philosophical Society. Vol. 48, part ii.

clearly illustrated by the following figures (Table X.) taken from Wohler's results.†

Attachment of Blades.—There are a great many possible methods of attaching the blades to the rotor or casing. In the Riedler-Stumpf turbine the moving blades are cut out of the solid rim of the wheel. In Curtis turbines the blades are cut from solid rings—or segments—of steel, which are then bolted on to the discs or wheels or fitted into grooves in the casing. The blades of a De Laval turbine are dovetailed into the rim, and those of the Rateau turbine are riveted on to the flanged rim of the disc. The blades of the Zoelly turbine have two small lugs, one on either side at the inner end, and are clamped between two plates which form the disc or wheel. The method usually adopted by builders of the Parsons turbine has already been described as also the method adopted by the Willans and Robinson Company. One method, not now in actual use the writer believes, consisted in electrically welding the blades to the rim of the drum, wheel, or steel (casing) ring. Soldering and binding on with wire have also been adopted. It is desirable that whatever be the method adopted the blades should be capable of removal for purposes of renewal; and the nature of the construction should be such that repairs can be carried out with the appliances usually available at a power house or factory.

TABLE X.

Material.	Stress, tons per square inch, tension.				Repetitions.
	Maxm.	Minm.	Mean.	Range.	
Homogeneous iron	19·10	0	9·55	19·10	34,500,000*
	33·41	14·31	23·86		1,234,600
	38·20	19·10	28·65		475,500
Spring steel, not hardened ...	28·65	0	14·3	28·65	468,000
	33·41	4·77	19·1	28·64	286,000
	38·20	9·55	23·9	28·65	176,300
	42·95	14·33	28·7	28·62	156,200
Spring steel, hardened	38·2	0	19·1	38·2	339,150
	57·3	19·1	38·2		35,600

† See also, "A possible explanation of the phenomena caused by repetitions of stress," *Mechanical Engineer*, November 22nd and 29th, 1902.

* Not broken.

In Fig. 103 we have illustrated the usual method of attaching the blades to either the rotor or casing of a reaction turbine. The groove turned in the rotor or

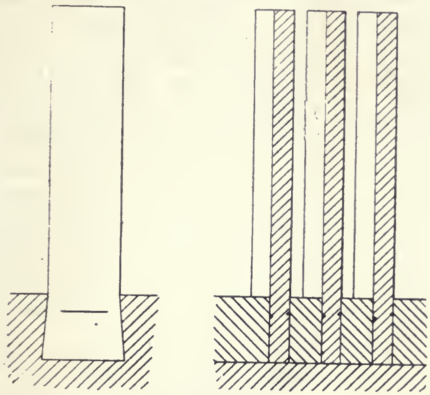


FIG. 103.—ATTACHMENT OF BLADES OF A REACTION TURBINE.

casing is slightly narrower at the top than the bottom. The caulking or filling pieces, which fill the grooves, between consecutive blades, are of soft brass or steel cut

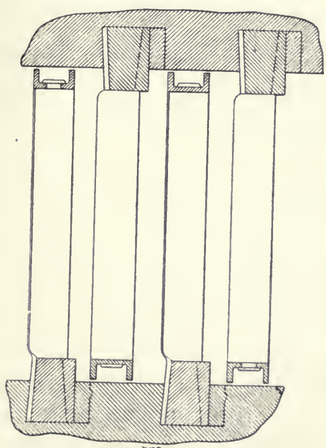


FIG. 104.—BLADES OF REACTION TURBINE.

from rolled or drawn rods of the correct section. The blades are sometimes nicked † as shown in the figure, so that the caulking pieces will have a firmer grip on them.

† More generally slight corrugations are pressed on.

The caulking spreads the caulking pieces and to some extent the blade possibly, so that they completely fill the groove The blades shown in Fig. 104 are fitted into a brass foundation ring at the bottom, and have a channel shroud riveted over their ends. A brass caulking strip binds the foundation ring firmly in the groove. Fig. 105 shows the method of attaching the fixed and moving blades in a Curtis turbine. The fixed blades are usually cut from half rings or shorter segments in the case of very large turbines, and fit into shallow grooves turned in the casing, being further secured by bolts as shown. The moving blades are usually cut in complete rings, although sometimes short segments only are used, which facilitates renewals and to some extent the cutting of the blades.

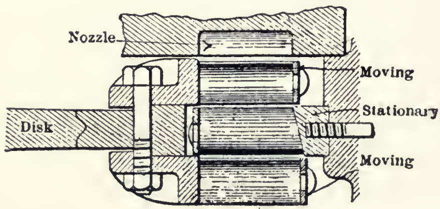


FIG. 105.—ATTACHMENT OF BLADES OF CURTIS TURBINE.

These rings have a slight shoulder turned on the radial face which bears against the inside of the rim turned on the wheel, thus insuring a perfectly central adjustment of the blades. As a rule there are two such rings, one on either side of a wheel, and fastened to the wheel by bolts passing through the rings and the rim of the wheel.

In Figs. 22, 23, and 25 we saw how the blades of Zoelly turbines were held in place. The moving blades have two small lugs, or an enlargement of some sort on the inside end. This enlargement fits into a groove formed either by the two halves of the disc—each disc being sometimes really two discs clamped together—or the rim of the disc and a clamping ring. The former method is illustrated in Fig. 24, and the latter in Fig. 22. The blades themselves are machined out of nickel steel, and polished in order to reduce the friction. As a rule, the thickness of the blade (Fig. 25) diminishes as we approach the outer end, so as to reduce the centrifugal stress at the root of the blade. The distance-pieces inserted in the grooves

between the blades are shaped to give (at the low-pressure end) a greater blade depth on the outlet than on the inlet side. The fixed blades are attached in a rather different manner. A wrought-iron ring or tyre fits over the ends of the blades, being screwed on to the distance-pieces *P*, Fig. 23. The blades, usually of sheet steel, have two small ears on their low-pressure (exhaust) side, which fit into corresponding recesses in the diaphragm and tyre. Two rings are then screwed on to the low-pressure side of the diaphragm and tyre, and thus hold the blades firmly in position. These diaphragms are in halves, and fit into shallow grooves in the casing, the upper half being held to the top half of the casing by set-screws so as to lift with the casing. Fig. 106 shows another method of attaching the blades of an impulse turbine to the discs.

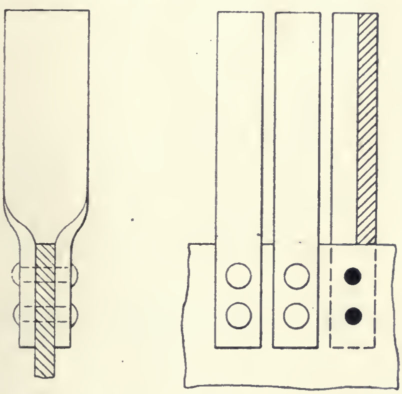


FIG. 106.—ATTACHMENT OF BLADES OF AN IMPULSE TURBINE.

Blade Lashings.—Long blades require some support near their outer ends. This is supplied by the shrouding, where fitted, but as most reaction turbines have no shrouding some form of "lashing" has to be provided. The usual method of applying such lashing is illustrated in Fig. 107. A slot is cut or punched out of the outlet edges of the blades and a wire, usually of rectangular cross section, is laid in the slots. This is then bound on to each blade by fine wire somewhat as shown in the figure, and soldered with silver solder. Ordinary solder is too soft, and the high temperatures used with the silver soldering process frequently result in burnt blade ends.

The process is naturally slow and expensive, and in order to obviate its use various mechanical lashings have been suggested. Fig. 108 shows a method devised by the author. A *V* slot is cut out of the blades and a strip of

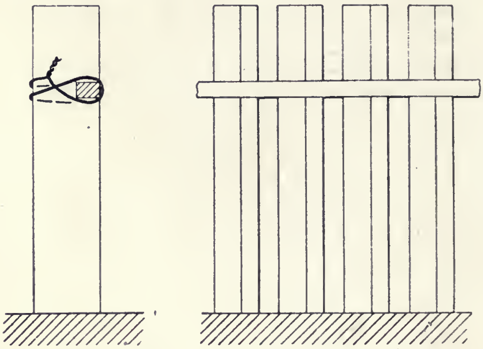


Fig. 107.—Usual Type of Blade Lashing for Reaction Turbine.

metal previously folded nearly flat on itself down the centre line is forced into the slot with one-half of the strip in each leg of the *V*. The two halves of the strip between the blades can then be pressed together by

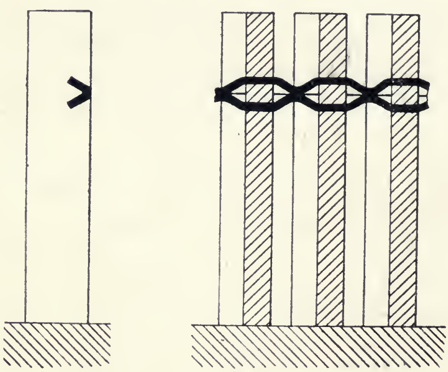


Fig. 108.—Suggested New Blade Lashing.

means of a pair of specially-shaped pliers, and the lashing is complete.

With quite long blades it is sometimes found that the extra expansion of the circumferential supporting strip above that of the steel drum causes the blading to buckle.

In order to prevent this the firm of John Brown & Co., of Clydebank, divided the strip into segments, alternately of solid rod and tubing. The ends of the rods fitted into the tubing, so that a continuous strip with several expansion joints was obtained.

It is found to be quite impossible in ordinary practice to tell beforehand with any degree of accuracy the number of blades which will be required to fill a given groove, even though the same size of blades and groove have been used before. This is undesirable, and is one more argument in favour of the adoption of some system of setting up the blades by machinery. Fig. 109 shows the kind of spacing that is quite common in reaction turbine blades. It will be seen that although the total inlet and outlet passage areas are not altered, yet the cross-section midway between the blade edges is very seriously affected.

The spacing of the blades in consecutive rows is sometimes purposely varied so as to give an increasing passage

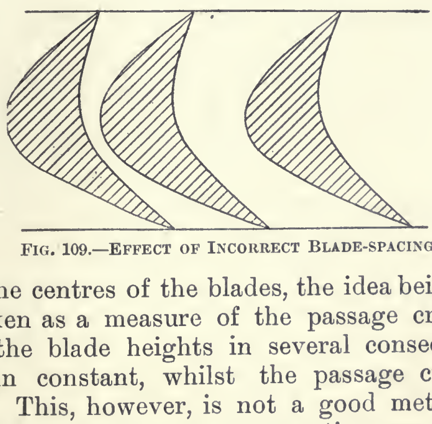


FIG. 109.—EFFECT OF INCORRECT BLADE-SPACING.

width at the centres of the blades, the idea being that this can be taken as a measure of the passage cross-section, and thus the blade heights in several consecutive rows will remain constant, whilst the passage cross-section increases. This, however, is not a good method, as the entrance and outlet passage cross-sections are not altered, and they are of greater importance than the mid-blade passage cross-section.

Gauging Factor.—The ratio of the width of the steam path between blades—measured at right angles to the mean line of flow—at inlet to that at outlet is called the gauging factor. Its value in Parsons turbines is usually about 3, but is reduced to about $2 \cdot 5$, and even as low as 2 in very long blades. Except for quite small blades

(in which the clearance leakage would be excessive) the author is of the opinion that a smaller gauging factor, something like 2, would give better results.

The blade heights in a reaction turbine never follow the theoretical values exactly. Instead, the height will be maintained constant for several consecutive rows, the number of rows in such a sub-section being greatest at the high-pressure end. The error thus introduced is small so long as the actual blade height does not vary by more than about 10 per cent. from the theoretical value. The error, however, can be reduced by decreasing the gauging factor as the pressure falls in one of these sub-sections of constant blade heights. This means that a blade having normal inlet and outlet angles of, say, 70° and 30° would be placed in the grooves in such positions—which are determined by the caulking pieces—as to make the angles change from, say, 75° and 25° at high-pressure end of the sub-section, to 65° and 35° at the low-pressure end, thus reducing the gauging factor from 2·26 to 1·58. By this means the blade angles can be kept at practically the theoretical values.

This method is not followed extensively, although the author understands it is sometimes used. A more detailed description of its theory will be given in Chapter XIII.

As far as practicable, however, the blade heights should correspond with the theoretically best values—when allowance has been made for losses.

CHAPTER V.

ROTORS.

Length of Rotor.—The length of the rotor is a matter of great importance owing to the difficulty of securing perfect rigidity with long rotors. Although within limits the power of the turbine does not affect the number of stages, because the work done in each stage depends on the peripheral velocity and blade angles only, yet it does affect the length. The longer blades required in large turbines must be wider axially and have larger clearances than the shorter blades of the low-power turbine. Fitting a shroud or supporting ring over the ends of long blades enables narrower blades to be used, and hence also a somewhat shorter rotor. These supports also enable the clearance between fixed and moving blades to be less because of the reduced chances of a blade getting out of alignment.

The length of the rotor can also be shortened by dividing the total expansion of the steam among two or more cylinders. This becomes almost a necessity in very large impulse turbines of the many-stage class. It has been tried in some of the larger Parsons turbines, but was found to be an unnecessary complication, and is never resorted to now except in marine work.

In a Parsons turbine the work done in each stage can be increased, and hence the number of stages decreased by making the inlet angle of the blades less than a right angle. This is illustrated by the following table. It will be noticed how rapidly the work done increases as the inlet angle gets smaller. The work done with angles of 70 and 40 degrees is 45 per cent. greater than with an inlet angle of 90°, the outlet angle having any value.

The inlet and outlet angles must not, however, be too nearly alike. For reasonably large turbines the best

angles are probably those previously determined from consideration of the friction loss.

TABLE XI.

Outlet Angle β	50	40	30	20
Inlet Angle α				
90	100	100	100	100
80	127	118	111	107
70	170	145	127	115
60	325	197	150	127
50	...	350	195	143
40	...	...	323	174

This question of the rotor length is of very great importance in marine turbines, where, owing to the low peripheral speeds adopted—seldom as high as 200ft. per second—the number of stages is very large. Such turbines are usually subdivided into several cylinders.

The following table shows the relative number of stages required for different peripheral speeds, the number being taken as 100 per cent. for a peripheral speed of 1,000ft. per second.

Feet per second ..	1,000	500	300	200	150	100
Number of stages per cent.	100	400	1,110	2,500	4,450	10,000

A reduction of the number of stages below the calculated number does not cause a proportionate reduction in the efficiency of the turbine. Consequently, where first cost and size are of primary importance—as in electrical generating stations with small load factors—a small percentage of thermodynamic efficiency may be sacrificed in order to simplify the construction.

Since at light loads (even when using a blast governor) the initial steam pressure is very much less than the boiler pressure, it follows that in order to secure a high efficiency the number of stages at these light loads must be considerably reduced. It being impracticable to

vary the number of stages with the load, the turbine must be designed with relatively few stages.

Discs and Drums.

The stresses in a rotating plate disc of uniform thickness are greatest at the centre. Making a hole at the centre for the shaft weakens the disc considerably. By thickening the disc about the centre we can strengthen it, and thus increase the safe speed at which the disc may be run. For these high-speed discs two things are desirable; at no place in the disc should the stress exceed a certain limit, and the greatest stress should be at the rim, so that should the disc run away the only result will be to break off the blades, which will relieve the disc of some stress. If the greatest stress is at or near the centre then should the disc run away rupture will start at the centre and the whole disc will go to pieces, with probably disastrous results. It is specially desirable that the greatest stress be at the rim, because it is possible to determine it there with fair accuracy; whereas near the centre the great width of the disc and the influence of the fastenings make the stresses somewhat uncertain. External to the rim of the disc proper we have the blades and a thicker ring of metal to which they are attached. This ring of metal is apt to have a disturbing effect on the stresses in the disc proper by relieving it of some of the centrifugal loading due to the blades and this ring. This can be got over by slitting the ring radially at several points. This has the further advantage that sudden variations in the steam temperature, such as occur when starting the turbine in an emergency, will not induce unknown stresses in the discs due to the expansion or contraction of this ring under the influence of the varying temperature.

There are two kinds of stress in a rotating disc: a radial stress (p) and a tangential stress (f). These are produced by the centrifugal force due to the disc proper and that due to the non-self-supporting portion of the disc, which is external to the rim of the disc proper.

In our calculations we use pounds, feet, and seconds, or grammes, centimetres, and seconds, as units of force,

length, and time. When the latter units are used, g is 981, otherwise $g = 32{\cdot}2$.

Then*

r = radius at any point.
b = thickness (at this point).
f = tangential stress.
p = radial stress.
ρ = density of material of disc (pounds per cubic foot or grammes per cubic centimetre).
$m = \frac{1}{\sigma}$ = Poisson's ratio $= 0{\cdot}248$.
e = base of Naperian logarithms $= 2{\cdot}718$.
E = modulus of direct elasticity.
C = external centrifugal loading due to the blades and the external ring for a length of periphery equal to the radius.
W = angular velocity = $6{\cdot}283$ times the revolutions per second.
u = radial displacement at any point due to stress.

Let the suffixes 1, 2, and 0 refer to the outer radius, inner radius (where there is a hole in the disc), and centre respectively.

Then remembering that each stress will produce a corresponding strain, and superposing these strains, we have

$$\text{Tangential strain} = \frac{2\pi u}{2\pi r} = \frac{u}{r}$$

$$\text{Radial strain} = \frac{du}{dr}$$

$$\text{Hence } \frac{Eu}{r} = f - mp \quad \text{. . . . . (1)}$$

$$E\left(\frac{du}{dr}\right) = p - mf \quad \text{. . . . . (2)}$$

Combining equations (1) and (2), we get

$$f = \frac{E}{1-m^2}\left(\frac{u}{r} + m\frac{du}{dr}\right) \quad \text{. . . . . (3)}$$

$$p = \frac{E}{1-m^2}\left(\frac{mu}{r} + \frac{du}{dr}\right) \quad \text{. . . . . (4)}$$

* Part of these calculations were published by the Author in the "Engineer" for January 8th, 1904.

Now consider the stresses acting on an elementary layer of the disc (Fig. 110) we have—

$$(p+\delta p)\ (r+\delta r)\ (b+\delta b) + \frac{1}{g}\, b\, r\, \rho\ W^2 r\, \delta r = p\, b\, r + f\, b\, \delta r$$

Simplified, this is

$$p\, b\, r + p\, b\, \delta r + p\, r\, \delta b + r\, b\, \delta p + \frac{\rho}{g} b\, r^2\, W^2\, \delta r = p\, b\, r + f\, b\, \delta r$$

Therefore $\delta\,(p\, b\, r) + \dfrac{\rho}{g}\, W^2 r^2\, b\, \delta r = f\, b\, \delta r,$

We may write this—

$$f\, b = \frac{d\,(p\, b\, r)}{dr} + \frac{\rho}{g} W^2 r^2 b \quad . \quad . \quad . \quad . \quad (5)$$

The total centrifugal force at any radius (on an arc equal to the radius) is—

$$F = \int \frac{\rho}{g}\, b\, W^2 r^2\, dr + C$$

but $F = \int f\, b\, d\, r + p\ br.$

Hence at the outer radius where—

$$\int f\, b\, dr = \int \frac{\rho}{g} b\, W^2 r^2\, dr$$

we have $$C = p_1\, b_1\, r_1$$

or $$b_1 = \frac{C}{p_1\, r_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

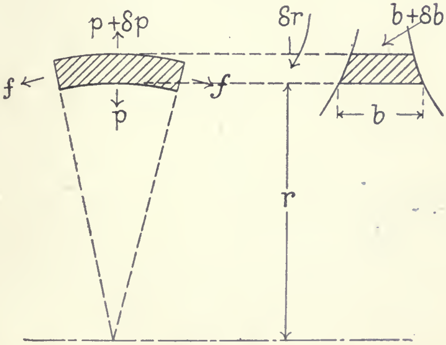


FIG. 110.

Solid Disc.—First let us consider the case of a disc without a hole in the centre. Such discs are much stronger than those with a central hole.

For simplicity in our calculations we shall denote

$$\frac{1 - m^2}{E} \text{ by } n$$

Let us assume a linear variation of stress and proportion the thickness to give this result.

Let $$f\,n = k + l\,r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

k and l being constants.

From equation (3) we have—

$$\frac{d\,u}{dr} + \frac{\sigma\,u}{r} = \sigma\,(k + l\,r).$$

The integrating factor for this equation is r^{σ}, so that

$$\frac{d\ (u\ r^{\sigma})}{d\,r} = \sigma\,k\,r^{\sigma} + \sigma\,l\,r^{\sigma+1}$$

or

$$u\,r^{\sigma} = \frac{\sigma\,k\,r^{\sigma+1}}{\sigma+1} + \frac{\sigma\,l\,r^{\sigma+2}}{\sigma+2}$$

No additive constant being necessary in this case, since $u = o$ when $r = o$, hence we have

$$\begin{cases} \dfrac{u}{r} = \dfrac{k}{\sigma+1} + \dfrac{\sigma\,l\,r}{\sigma+2} \\[2ex] \dfrac{du}{dr} = \dfrac{\sigma\,\mathrm{k}}{\sigma+1} + \dfrac{2\,\sigma\,l\,r}{\sigma+2} \end{cases}$$

Substituting these values in equation (4) we have

$$p\,n = k + l\,r\left(\frac{1+2\sigma}{\sigma+2}\right) = k + q\,r \quad . \quad . \quad . \quad (8)$$

Substituting these expressions for f and p in equation (5) we obtain

$$\frac{d\,b}{d\,r} + b\left[\frac{2\,q - l + \dfrac{\rho}{g}\,n\,W^2\,r}{k + q\,r}\right] = o$$

This is the form

$$\frac{d\,b}{d\,r} + P\,b = o$$

$$P = \frac{2\,q - l + \dfrac{\rho}{g}\,n\,W^2\,r}{k + q\,r} = \frac{d_1 + a_1\,r}{k + q\,r} = \frac{d + a\,r}{r + w}$$

$$= a + \frac{h}{r+w} \text{ where } h = d - a\,w$$

The integrating factor is $I = e^{\int P\,dr} = e^{a\,r}\,(r+w)^h$

Hence we have

$$b\,I = \text{a constant}$$
$$= b_1\,I_1$$

and

$$b = b_1\left(\frac{I_1}{I}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Thus all that is necessary to design the disc is to fix on the stresses (either f or p) desired, and from these to calculate the values of the various constants k, l, q, &c., after which a little rather tedious arithmetic gives us the values of I at various radii; b_1 is obtained from equation (6), and all the other values for b at the various radii are obtained at once from equation (9).

No knowledge of higher mathematics is required at all, and those who want can skip all the intermediate processes in the above calculations and just insert values in the equations given.

For example we will assume that

$r_1 = 3$ft.
$W = 300$ (2,870 revs. per minute).
$C = 0{\cdot}022 \times 10^6$lbs. (3·28 tons per foot of rim).
$f_1 = 4 \times 10^6$lbs. per square foot (12·4 tons per square inch).
$f_0 = 3 \times 10^6$lbs. per square foot (9·3 tons per square inch).
$n = 2{\cdot}16 \times 10^{-10}$
$m = 0{\cdot}248$.
$\rho = 490$lbs. per cubic foot.

Then we find that

$l = 0{\cdot}72 \times 10^{-4}$
$k = 6{\cdot}48 \times 10^{-4}$
$q = 0{\cdot}48 \times 10^{-4}$
$d = 0{\cdot}5 \times 10^{-4}$
$a = 6{\cdot}17$
$w = 13{\cdot}5$.
$h = -83{\cdot}3$.
$p_1 = 3{\cdot}67 \times 10^6$ (11·36 tons per square inch).
$p_0 = 3{\cdot}0 \times 10^6$

and we have the following values for b in inches at different radii, r being in feet.

r	3	2·5	2	1·5	1	0·5	0·0
b	0·24	0·42	0·635	0·9	1·18	1·36	1·425

The above values have not been calculated with a very great degree of accuracy, such as is desirable in practical work, as it is easy to make a considerable error

in the value for the thickness near the centre, where fortunately it is not of any very great consequence. The stress at the centre should not be much less than 0·75 that at the rim, or the disc will become too thick at the centre. Since the stresses at the centre are not greatly affected by a moderate change in thickness, it follows that the presence of collars or other projections for fastening the disc to a shaft will not have any very serious effect on the stresses.

Disc with a Central Hole.—The method of procedure is similar to that for a solid disc.

If $$f\,n = k + l\,r,$$

we have $$u\,r^{\sigma} = \frac{\sigma\,k\,r^{\sigma+1}}{\sigma+1} + \frac{\sigma\,l\,r^{\sigma+2}}{\sigma+2} + K,$$

K being a constant due to the fact that the strain u at the centre is not zero.

Proceeding as before we obtain

$$p\,n = k + l\,r\left(\frac{1+2\sigma}{\sigma+2}\right) + \frac{K}{r^{\sigma+1}}\left(\frac{1-\sigma^2}{\sigma}\right).$$

Now, when $r = r_2$, $p = 0$, so that we obtain the value of

$$K\left(\frac{1-\sigma^2}{\sigma}\right) = -\,D,$$

which makes

$$p\,n = k + q\,r - \frac{D}{r^{\sigma+1}}$$

Substituting the values of f and p in equation (5), we get an equation of the form

$$\frac{d\,b}{d\,r} + P\,b = 0$$

where

$$P = \frac{\frac{\rho}{g}\,n\,W^2\,r + 2\,q - l + \frac{\sigma D}{r^{\sigma+2}}}{k + q\,r - \frac{D}{r^{\sigma+1}}}.$$

As before, the integrating factor $I = e^{\int P\,dr}$ and

$$\frac{b}{b_1} = \frac{I_1}{I}$$

The simplest way to obtain I is to plot a curve between P and r somewhat as in Fig. 111. Then the area under

this curve from $r = r_2$ up to the point considered gives us the value of $\int P\,dr$. Just as we near r_2 the curve goes off to infinity; take some radius as close to r_2 as possible consistent with being able to determine the exact position of the curve there. Then if a is the area of the curve from this radius we have

$$\frac{b}{b_1} = \frac{e^c}{e^a}$$

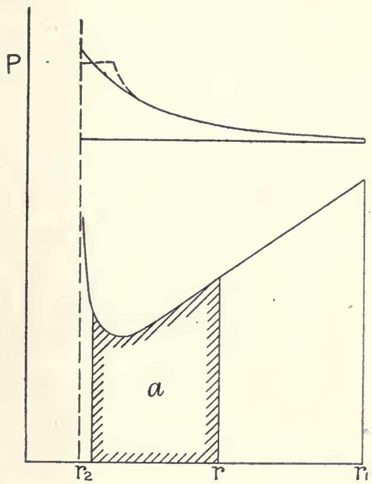


FIG. 111.

where c is the area up to the external radius and a up to the radius corresponding to b. In this way we obtain the shape of the disc down to a radius very little different from r_2. In order to determine the thickness at r_2 we must try and obtain an empirical formula between b and r which will enable us to calculate b_2. In practice we cut off a little bit of the final thickness b_2 and increase b for some little greater radius as is indicated in Fig. 111. Care is required to insure that we thicken up sufficiently.

When using a disc with a central hole it is good practice to thicken up at the hub as much as possible. This is well illustrated by Figs. 22 and 26, the hubs of consecutive discs being widened until they meet, the diaphragms fitting over these hubs, and not directly over the shaft.

A rather better stress distribution than the above would be obtained if

$$f\,n = \frac{k_1}{r} + k_2\,r^2,$$

k_1 and k_2 being constants suitably chosen in order to make f_2 about 20 per cent. less than f_1. In this case we again obtain an equation of the form

$$\frac{d\,b}{d\,r} + P\,b = o$$

and hence

$$I = e^{\int P\,dr}$$

and

$$\frac{b}{b_1} = \frac{I_1}{I}$$

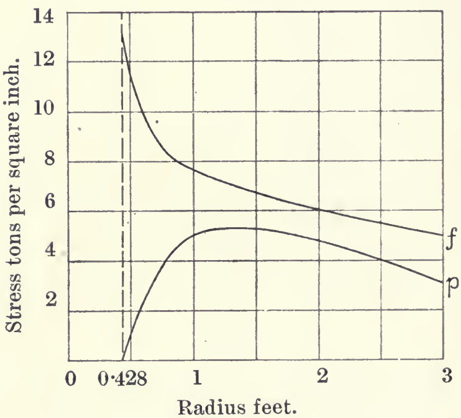


FIG. 112.—STRESSES IN A PLATE DISC.

now

$$p\,n = \frac{q_1}{r} + q_2\,r_2 - \frac{D}{r^{\sigma+1}}$$

and $p = o$ when $r = r_2$, which enables us to find D. Also

$$q_1 = \frac{k_1}{\sigma} \text{ and } q_2 = k_2\left(\frac{3\,\sigma + 1}{\sigma + 3}\right)$$

and

$$P = \frac{\frac{\rho}{g}\,(n\,W^2\,r^2) + \frac{\sigma\,D}{r^{\sigma+1}} - \frac{k_1}{r} + r^2\,(3\,q_2 - k_2)}{q_1 + q_2\,r^3 - \frac{D}{r^{\sigma}}}$$

Plate Disc of Uniform Thickness.—Since b is constant equation (5) becomes—

$$f = \frac{d\,(p\,r)}{d\,r} + \frac{\rho}{g} W^2 r^2$$

Inserting the values of f and p as given by equations (3) and (4) this becomes—

$$r\frac{d^2 u}{d\,r^2} + \frac{d\,u}{d\,r} - \frac{u}{r} + \frac{n\,\rho\,W^2\,r^2}{g} = o$$

Integrating we get—

$$\frac{u}{r} = K_1 + \frac{K}{r^2} - \frac{n\,W^2\,r^2\,\rho}{8\,g}$$

$$\frac{d\,u}{d\,r} = K_1 - \frac{K}{r^2} - \frac{3\,n\,W^2\,r^2\,\rho}{8\,g}$$

Inserting these values in equations (3) and (4) we have

$$f\,n = K_1\,(1 + m) + \frac{K\,(1 - m)}{r^2} - \frac{(1 + 3\,m)\,(n\,\rho\,W^2\,r^2)}{8\,g}.$$

$$p\,n = K_1\,(1 + m) - \frac{K\,(1 - m)}{r^2} - \frac{(m + 3)\,(n\,\rho\,W^2\,r^2)}{8\,g}$$

For example

$n = 2{\cdot}16 \times 10^{-10}$
$m = 0{\cdot}248.$
$\rho = 490$ lbs. per cubic foot.
$p_1 = 10^6$lbs. per square foot (3·1 tons per square inch).
$r_1 = 3$ft.
$r_2 = 0{\cdot}428$ft.
$W = 141{\cdot}4$ (1,350 revs. per minute).
$p_2 = 0.$

Inserting the values of p at r_1 and r_2 we obtain K, and from this also K_1. From these data we can then calculate f and p at any radius. Fig. 112 shows the distribution of stress in the disc, the greatest stress being a tangential one at the inner radius. We obtain

$K = 1{\cdot}12 \times 10^{-4}$
$K_1 = 3{\cdot}725 \times 10^{-4}.$
$f_1 = 1{\cdot}6 \times 10^6$ (4·96 tons per square inch).
$f_2 = 4{\cdot}27 \times 10^6$ (13·24 tons per square inch).

The peripheral velocity of the above disc is 450ft. per second. The thickness of the disc depends on the

centrifugal loading (C) caused by the blades and their attachments.

Thus if C is $0{\cdot}15 \times 10^6$ over an arc equal to the radius (22·3 tons per foot of rim) the thickness of the disc will be 0·6in. If W is the weight of the blades and attachments per foot of rim and v their linear velocity in feet per second, we have

$$C = \frac{W\,v^2}{g} \ldots . \text{(over arc equal to radius)},$$

which in this case corresponds to 3·5lbs. per foot of rim for the weight of these attachments. W must include all that portion of the rim which is external to the disc proper, and which is used for making fast the blades.

For the highest speeds a 10 per cent. nickel steel having a tensile strength of 127,000lbs. per square inch and an elongation of about 12 per cent. is recommended by Krupp's. For medium speeds a good forged-steel disc with a medium carbon contents is suitable. Saw blades—without the teeth—are sometimes used when a plate disc is desired. The discs should be accurately machined all over so that any flaws may be discovered; to reduce the disc friction and to assist in obtaining a good balance.

Drums.—We shall assume that the stress due to centrifugal force is uniformly distributed over that portion of the surface carrying each set of moving blades. For instance, on either side of a blade 1in. wide there will be a length of drum equal to say 1in., which may properly be considered to be within the territory of the blade. We shall assume the stress over the whole 3in. length of drum to be uniform. This will give a stress somewhat lower than the maximum.

Consider 1in. axial length of drum.

Let a = area of the effective cross-section in square inches (equal to the mean thickness of the "solid" portion of the rim).

w = weight in pounds of the rim and all attachments per *foot* of rim.

v = rim velocity in feet per second.

f = tangential stress in pounds per square inch.

Then

$$f = \frac{w\,v^2}{a\,g}$$

and is independent of the radius for a given peripheral velocity. The same formula obviously holds true for any other axial length of drum than 1in., and it is usually convenient to consider the whole territory of one set of blades, but in this case a is not the mean thickness of the rim. The weights of the solid portion of the drum and of the attachments (blades, fastenings, &c.), can be estimated separately. The weight of the solid or effective rim when of steel is (per foot of periphery) :—

$$3 \cdot 4 \ a \text{ lbs.}$$

The weight of the attachment is easily found. In the case of a Parsons turbine we divide this weight into two portions ; the external portion of the blade and that portion, including caulking pieces, which fills the circumferential groove. For example, suppose the territory of a set of moving blades to be 2·4in. measured axially (Fig. 113), there being 20 blades per foot of rim each weighing—without that portion in the groove—0·15lbs.,

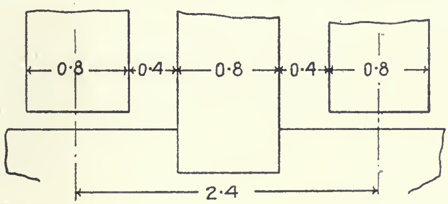


FIG. 113.—WIDTH OF TERRITORY OF ONE SET OF MOVING BLADES.

making a weight due to this portion of the attachments of 3lbs. Owing to the fact that the outer end of the blade has a velocity about 10 per cent. in excess of v (the drum speed), this weight ought to be increased about 10 per cent., making an " equivalent " weight due to this cause of 3·3lbs. per foot. If the groove is 0·8in. wide by 0·3in. deep and the material with which it is filled weighs 525lbs. per cubic foot, the weight of the groove material will be 0·875lbs. per foot. This makes the total weight of the attachments 4·2lbs. per foot. If the mean effective thickness of the rim is x inches, the weight of the effective portion of the rim is $8 \cdot 16 \, x$ lbs. per foot.

If we allow a mean stress of 8 tons per square inch, then we have

$$17{,}920 = \frac{(4 \cdot 2 + 8 \cdot 16 \, x) \, v^2}{2 \cdot 4 \, x \times 32 \cdot 2}$$

If $v = 350$ft. per second this will make x (the mean effective thickness of the rim) equal to 1·34in.

We have yet to consider the bending and twisting stresses in the drum. In the case of a broad wheel carried on a shaft extending between the bearings there will be no other stresses (of any great moment) than those due to centrifugal force, so that the above calculation would apply without any modification or addition.

This centrifugal force will slightly stretch the drum circumferentially and thus reduce the clearance. On the other hand the steam pressure will to a certain extent counterbalance this effect by expanding the casing and contracting the drum (circumferentially). The pressure effect is not very large, especially at the low-pressure end, where, indeed, the casing will be slightly contracted. Neglecting the effect of the steam pressure and the expansion due to temperature the radius of the drum will increase 0·0075 per cent. for each ton per square inch centrifugal stress. If the radius of the drum is 3ft. and the stress 10 tons per square inch the radial clearance will be reduced by 0·027in. The relative expansions of the drum and casing due to temperature are not so definite.

If t_1 = temperature of rotor above that at the time of construction.

t_2 = temperature of casing above that at time of construction.

0·000006 = linear coefficient of expansion of the casing—cast iron—per degree Fah.

0·0000067 = coefficient of expansion of rotor—steel.

r = radius in inches.

Then the expansion of rotor radially is $0{\cdot}0000067\, t_1\, r$ inches, and that of the casing is $0{\cdot}000006\, t_2\, r$ inches.

As a further refinement we ought, when the blades are long, to take account of the expansion of the blades and the difference in the radii of the rotor and casing. For instance, suppose that near the exhaust end of the turbine the radii of the casing and rotor are respectively 34in. and 30in., the blades being 4in. long. The temperatures of the casing and rotor we will suppose to be 170° and 200°, the temperature of construction being 60°, thus making t_1 and t_2 respectively equal to 140° and 110°. The

temperature rise of the blades will also be 140°, and if they are of brass with a coefficient of expansion equal to 0·0000103, we have

expansion (radial) of blades $= 0 \cdot 0058$in.
,, ,, drum $= 0 \cdot 028$in.
,, ,, casing $= 0 \cdot 0226$in.

The reduction in the radial clearance due to temperature is therefore

$$0 \cdot 028 + 0 \cdot 0058 - 0 \cdot 0226$$
$$= 0 \cdot 011 \text{in.}$$

If in this case the centrifugal stress in the rotor is 8 tons per square inch, there will be a further reduction of the radial clearance of 0·018in. from this cause. As the steam pressure is practically atmospheric it will not affect the clearance, and thus we see that the net reduction in the radial clearance is 0·029in. The usual clearance for a reaction turbine—measured cold—would be about 0·06in. The relative expansion of the rotor and casing under the influence of temperature may be reduced by thoroughly lagging the casing, and might even be entirely avoided by steam jacketing. This, however, would be no advantage, as it is the radial clearance under steam which is the important factor, and this must not be reduced below a certain indefinite limit whatever the temperatures may be.

The effect of the steam pressure is to increase the radial clearance, except where the pressure is below atmospheric. For the purposes of calculating this pressure effect, it is usually correct to take the pressure inside the drum as being equal to the condenser pressure. For instance, suppose that the tangential stresses induced in the rotor and casing by the steam pressure at a point where the radius is 15in. to be 1 ton and 0·75 ton per square inch respectively. Then the moduluses of elasticity of steel and cast iron being 13,500 and 5,500 tons per square inch, the casing will increase its radius by 0 .002in. and the rotor radius will decrease by 0·0011in., thus giving an increase of the radial clearance from this cause of 0·0031in. Generally speaking, this effect of pressure is very small.

Another change in the radial clearance is that due to the straightening of the shaft at full speed. Whilst the ends

of the blades are being ground up (in the shops) to a cylindrical form by an emery wheel, the rotor is sagging slightly on account of its weight. When running at full speed the rapidity of the reversal of the direction of stress prevents the full extension—and, therefore, the full sagging—from being attained. This lifts the rotor body somewhat, and reduces the radial clearance at the top of the casing. Another serious item which affects the radial clearance is the "whipping" or vibration of the rotor.

In practice the amount of clearance allowed over the blade tips is usually settled by experimenting on the finished machine. The turbine is designed and erected with certain clearances. It is then run up to speed on the testing bed and afterwards opened up. If the blade tips have touched, the clearance at that point has to be increased. This touching may be due to whipping on the part of the rotor—bad balancing—but is generally due to distortion of the cylinder.

It is common practice to make the axial clearances on the sides of the blades each equal to half the axial width of the blades. When the turbine is running, the greater expansion of the rotor relative to the casing will decrease these clearances on the thrust block—usually the high-pressure side—and increase those on the side away from the thrust block. The sides here refer to the casing blades; just the reverse would be true of the rotor blades.

Clearly, then, if the clearance which is reduced when running is still large enough to be safe, the clearances which increase when under steam are too large, and could be reduced; which by reducing the length of the rotor would considerably reduce the cost of construction.

The method of estimating the relative expansion of the rotor and casing is quite simple, though not very accurate, owing to lack of data. There is a fall of temperature from the admission port to the exhaust on the one hand and the stuffing box at the high pressure end on the other. There is then a short portion at approximately uniform temperature from the stuffing box to the thrust block. The casing will be at a somewhat lower temperature than the rotor, and in all cases expansion must be measured from the thrust block. We may

illustrate by an example. The distances from the thrust block, the temperatures of the rotor and casing, and the expansions in inches, measured from the thrust block, are given in Table XII.

We see from the table that the relative expansion of the rotor and casing (calculated by assuming a uniform temperature gradient between two consecutive positions) is equal to 0·0037in. at the high-pressure stuffing box, to 0·04in. at the first high-pressure blades, to 0·112in. at the last low-pressure blades, and to about 0·115in. at the low-pressure stuffing box.

If, in order to allow for inequalities in the alignment of the blades and other causes, we allow (axially) from 0·06in. at the high-pressure blades, to 0·16in. at the low-pressure blades, then the clearances on the two sides of the blades should increase from 0·06in. and 0·1in. at the high-pressure end up to 0·16in. and 0·27in. at the low-pressure end. The exact amount to be allowed for the inequalities depends on the blade lengths and the type of blades used. By the adoption of these unequal axial clearances on the two sides of a blade it is often quite feasible to shorten the rotor by a foot. It may be objected that the temperature ranges assumed in Table XII. are too great; but it must be remembered that the design must allow for the

TABLE XII.

	Temp. Fah.	Temp. of Construction.	Rise of Temp.	Distance from Thrust Block.	Expansion in inches.	Relative Expansion in inches.
Thrust block, casing ...	80	60	20	—	—	—
,, ,, rotor......	110	60	50	—	—	—
Stuffing box, casing....	90	60	30	20	0·003	0·0037
,, ,, rotor......	110	60	50	20	0·0067	
Admission, casing	450	60	390	70	0·066	0·04
,, rotor	600	60	540	70	0·106	
Exhaust end, casing ...	90	60	30	170	0·192	0·112
,, ,, rotor......	110	60	50	170	0·304	

maximum possible relative expansion, and this is probably even greater than that calculated in the table. Turbines are sometimes run without any lagging, and at the same time using highly-superheated steam.

Bending Stresses in the Rotor.—The rotor is subjected to bending stresses due to its own weight. It is simpler and quite accurate enough to divide this weight into a fair number of concentrated loads. For instance, if the weight of 1ft. axial length of the rotor were 3,000lbs., then we should consider that 3,000lbs. as being situated at the middle of that foot length. In this way we very easily obtain a curve of bending moments for the rotor. Then if M is the bending moment in inch-pounds at any point and I the moment of inertia of the rotor cross-section there (dimensions in inches) the stress f in pounds per square inch at a radius r inches will be

$$f = \frac{M r}{I}$$

Of course, r must lie within the metal of the rotor. For instance, where the rotor consists of a shaft carrying several discs the maximum radius is equal to the radius of the shaft.

In calculating the moment of inertia of the rotor, the external diameter of the drum should not be that of the drum surface, but should be very little greater than the diameter at the bottom of the blading grooves.

A usual construction for the drum of a Parsons turbine consists of a long hollow cylinder (possibly carrying one or more wheels) into the ends of which two short shafts have been pressed. Provided that the press fit is sufficiently tight to prevent any slipping of the surfaces there, we may consider the two shafts and the cylinder as one unbroken piece of metal. As a rule, the wheels carried on the outside of this cylinder do not take any bending stresses.

Where the drum is carried on a shaft running the full length between the bearings, both the drum and shaft are subjected to bending stresses. Thus, referring to Fig. 114 the weight of the drum is carried by the shaft at the two points P_1 and P_2; the sum of the pressures exerted by the drum on the shaft at these points being equal to the weight of the drum. The shaft then is acted upon by its own weight, and these two loads. The drum, on the other hand, is now—so far as bending is concerned—a beam subjected to its own weight and carried

on two supports, P_1 and P_2. Having once determined the pressures on the shaft at P_1 and P_2 the drum and shaft can be considered quite independently.

Shearing Stresses due to a Twisting Moment on the Rotor.—The twisting moment on any set of moving blades can be calculated from the blade angles, the weight of steam passing per second, the steam and wheel velocities. For 1lb. of steam this is in a Parsons turbine

$$r\,\frac{(v^2_1 - v_2^2)}{V g} \text{ – pounds feet,}$$

and the twisting effort is $\dfrac{v^2_1 - v_2^2}{V g}$.

The twisting effort for an impulse turbine with symmetrical moving blades, the velocity at the outlet being equal to that at inlet, is

$$\frac{2}{g}\,v_2 \cos a.$$

The above refer to a single set of moving blades. Where the steam velocities or blade angles are not known we can readily calculate the twisting moments on each set of blades from the "indicated" power of the turbine. This will be best illustrated by taking an example as follows :—

4,000 i.h.p.
1,000 r.p.m.

33 rows of blades having a peripheral speed of 120ft. per second,
18 rows of blades with a peripheral speed of 250ft. per second,
10 rows of blades with a peripheral speed of 360ft. per second.

The total twisting moment is—

$$\frac{4{,}000 \times 33{,}000}{6{\cdot}28 \times 1{,}000} = 21{,}000 \text{ foot-pounds.}$$

The twisting moments on the individual rows of blades are in the ratio of the squares of their peripheral velocities. If $144m$, $625m$, and $1{,}296m$ are these moments, then we have

$$33 \times 144m + 18 \times 625m + 10 \times 1{,}296m = 21{,}000,$$

so that

$$m = 0{\cdot}725,$$

and the twisting moments per row at the different diameters are respectively 104, 453, and 940 foot-pounds. Starting at the high-pressure end the twisting moment will increase progressively until at the exhaust end of the drum it reaches the full value of 21,000 foot-pounds. We can easily draw in a curve of twisting moments as in Fig. 114. Where the drum is merely carried on a shaft

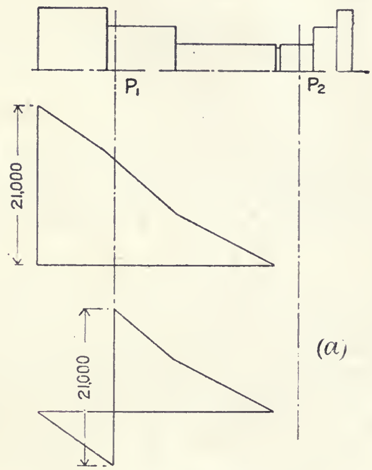


FIG. 114.—TWISTING MOMENT ON ROTOR.

the twisting moment curve will be somewhat as indicated in Fig. 114a, being zero at both ends, and at an intermediate point, but having the sum of the values at the points of support P_1 and P_2, equal to 21,000. This will also give us the twisting moments in the shaft. In the figure all the twisting moment is transmitted through the support P_1.

If at any point in the rotor we have a bending moment M and a twisting moment T (both in inch-pounds) then the equivalent bending moment is—

$$m = \frac{1}{2}\left[M + \sqrt{(M^2 + T^2)}\right]$$

from which the stress due to the combined bending and twisting moments may be calculated. It may perhaps be as well to mention here that if d_1 and d_2 are the external

and internal effective radii of the rotor in inches, the moment of inertia of the cross-section is

$$I = 0{\cdot}049\ (d_1^4 - d_2^4).$$

If at any point along the rotor the greatest stresses produced by this equivalent bending moment and centrifugal force are respectively f_1 and f_2, then the maximum normal or direct stress due to these two combined is half their sum. Hence, we calculate the thickness of the rotor metal for whichever stress f_1 or f_2 is the greater.

In the ordinary Parsons turbine rotor, the centrifugal stresses are comparatively small at the high-pressure end. The bending stresses, although quite small, usually limit the minimum thickness of the metal. At the low-pressure end of the rotor the centrifugal stresses are all important. At the stuffing boxes the combined twisting and bending stresses are the most important. They usually have a value between 5,000lbs. and 7,000lbs. per square inch, and are kept as low as possible in order to make the spindle stiff. For the same reason the bending stresses in the drum proper are kept very small, often not more than about 1,000lbs. per square inch.

Construction of the Rotor.—The rotors of impulse and reaction turbines differ fundamentally. In the impulse rurbine there is no fall of pressure across the blades of an individual wheel, and hence the radial clearance between the moving blades and the casing may be large, as there is no tendency for the steam to leak across. On the other hand, there is a fall of pressure between the two faces of the fixed blades, so that in order to prevent leakage past the diaphragm in which these blades are fixed, the radial clearance between the diaphragm and the rotor must be reduced to a minimum. These considerations have led to the adoption of a rotor consisting essentially of a stepped shaft carrying a number of individual wheels, separated from one another by fixed diaphragms extending from the casing to the shaft. In this manner the length and radial depth of the clearance space between the shaft and diaphragm is reduced to a minimum, and leakage is further reduced by the use of a labyrinth packing consisting merely of a number of small circular

grooves turned in the inner surface of the central hole of the diaphragm. Such a packing was illustrated in Fig. 13. A number of these rotors were illustrated in connection with the descriptions of the different types of

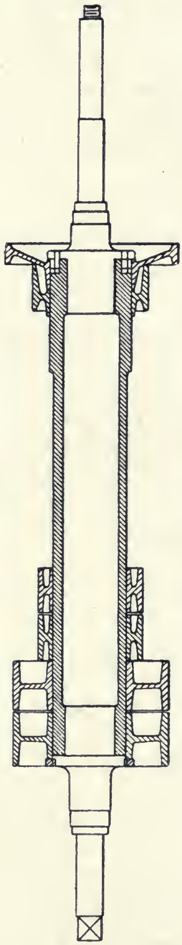


FIG. 115.—BUILT-UP DRUM ROTOR OF REACTION TURBINE.

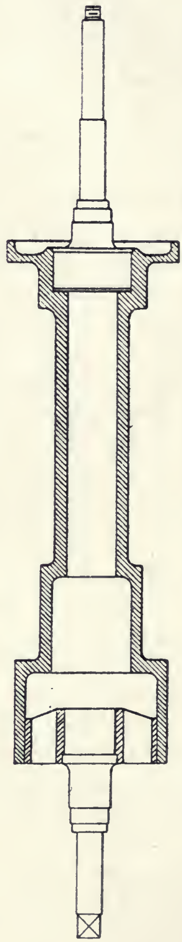


FIG. 116.—SOLID DRUM ROTOR OF REACTION TURBINE.

turbines given earlier on. Such a rotor is difficult to make stiff, but stiffness is not here a prime necessity.

It is otherwise with the rotor of a reaction turbine. There is in this case a fall of pressure across both fixed and moving blades, and hence in order to minimise

leakage very small radial clearances are necessary, and these cannot be secured without resource to a stiff rotor. This has led to the general adoption of a "drum" rotor or spindle. In Fig. 115 we have illustrated a very common type of such spindle. It consists of a forged steel drum of approximately uniform diameter carrying a number of cast-steel wheels for the purpose of stepping up the diameter where necessary. Two short forged-steel shafts are shrunk or forced into the ends of the drum, and further secured by bolts. The moving blades are mounted directly on this spindle, as was very clearly shown by Figs. 47 and 57. Fig. 116 shows another construction of spindle. In this case the drum proper is in one piece of forged or cast steel, usually the former. At the low-pressure end in the example shown a broad wheel is shrunk inside the drum. Two short bearing shafts are forced into the ends of this rotor. Fig. 117 shows a design of rotor for a large 2-cylinder tandem turbine.

In Fig. 50 we illustrated the rotor of a British Westinghouse turbine. This rotor has a through-going shaft. As a general rule this is not desirable in turbines with drum rotors, as the spindle can be made much stiffer for a given weight when the stresses are borne by the drum and not by a central shaft. In this case, however, the length between bearings is comparatively short.

As far as possible cast steel should be avoided in the construction of the rotor. It is much cheaper than forged steel, but it lacks homogeneity, and is consequently somewhat uncertain as to strength and difficult to balance. In any case, the rotor should be accurately machined both inside and out, in order to facilitate balancing and to discover any internal flaws in the metal. The rotor must, of course, be perfectly symmetrical with respect to its axis, both for balancing reasons and in order to prevent distortion by the heat.

In order to prevent condensation of the steam as it passes through the turbine, it is not advisable to allow the exhaust end of the turbine to communicate with the interior of the drum.

As regards the shrinking or forcing in of the end wheels and bearing shafts, we must remember that these

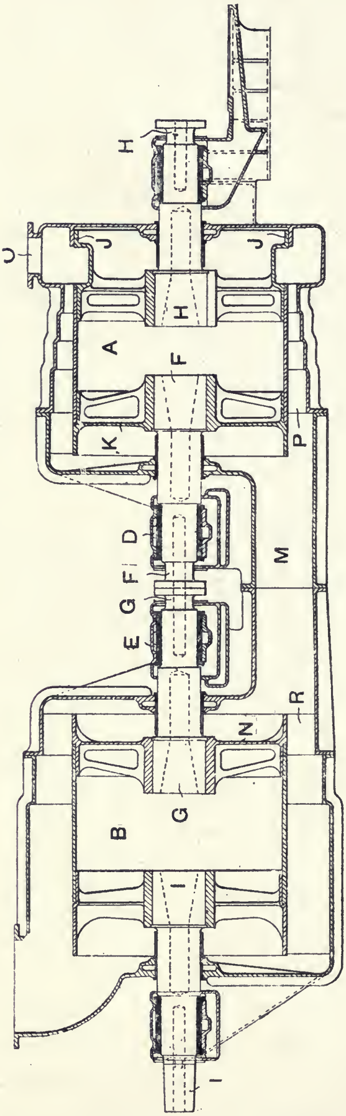


Fig. 117.—General Construction of Large 2-Cylinder Tandem Turbine.

joints have to transmit both bending and twisting stresses under running conditions. Frequently bolts

and keys are put in to relieve the joints of these stresses, but it is still necessary to insure that the joint be perfectly tight under running conditions.

For instance, suppose a shaft to be forced into an end-bell and this again to be shrunk into the drum. In most cases we cannot estimate with any accuracy the stresses which will be set up when running, so that it is best to assume that the shaft does not increase in diameter on account of centrifugal force, but that the end-bell does by an amount at least equal to that of a plain steel ring of the same diameter and peripheral speed as the shaft. Generally, it is best to assume considerably higher stresses. Suppose, then, we assume that the centrifugal stress in the end-bell at the hole to be 5 tons per square inch, and the diameter of the said hole to be 20in. Then the hole in the end-bell will increase in diameter by an amount equal to

$$\frac{\text{Stress}}{\text{Modulus of elasticity}} \times \text{diameter.}$$
$$= \frac{5}{13{,}500} \times 20$$
$$= 0 \cdot 0074\text{in.}$$

Then the end-bell must be bored out more than 0·0074in. smaller in diameter than the shaft in order to keep the joint tight when rotating. But as a rule this joint will also have to transmit bending stresses.

For instance, suppose the joint to transmit a bending moment M inch tons. This moment is balanced by a frictional force at the joint which varies proportionally with its distance from the diameter about which the moment is acting—if there is no slipping at the joint. If f tons per inch of circumferential joint is the maximum value of this force it can easily be shown by integration that

$$f = \frac{4\,M}{\pi\,d^2}$$

where d is the diameter at the joint in inches.

Then, since each part of the joint is called upon in turn to resist this force, the force with which the shaft

must be forced into the end-bell must be at least equal to $\pi\, df$ tons, that is to say

$$F = \frac{4\,M}{d}$$

For example, if in a large rotor the bending moment transmitted by the joint is 350in. tons, and the diameter of the joint 20in., the force required to make the joint must be at least 70 tons.

This force is only just sufficient to take care of the bending stresses.† In addition, therefore, we must have a force sufficient to prevent the centrifugal forces opening the joint by the expansion of the outer piece forming the joint. The twisting moment is usually taken care of by means of sunk keys. The shaft usually has a collar turned on it through which bolts are passedto bind it firmly against the end-bell or rotor. If necessary these bolts will then transmit the tension, and the collar itself the thrust, necessary to balance the bending moment M.

The calculations as to the force required to make a press fit are at the best not too reliable, so that it is always advisable to provide other means of transmitting the necessary forces across the joint.

The final machining of the rotor should be done when it is built up complete, otherwise there will always be some slight inaccuracies, however carefully it may be built up.

† Since a lubricant—usually white lead—is used when making a press fit, the actual force required to make the joint is less than that necessary to open it again.

CHAPTER VI.

GOVERNING.

THERE are four methods of varying the steam supply to suit the load on the turbine. They are :—

(1) Throttling the steam supply ;
(2) Varying the area of cross-section of the inlet steam passages ;
(3) Varying the time of admission ; and
(4) Admitting steam at various points along the direction of the steam flow.

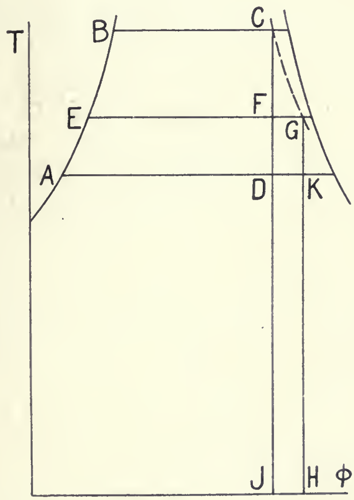


FIG. 118.—EFFECT OF THROTTLING.

Throttle Governing.—When steam is throttled its available energy decreases. It is true that no heat is lost, but it becomes degraded in temperature, and therefore the proportion of this heat which can be converted into work is reduced. For instance, we can obtain about twice as much work from a unit of heat in steam at 400° Fah. as

we could if its temperature were about 220°. Or again, if steam at 135lbs. (absolute) containing 2 per cent. of moisture is throttled to 65lbs. (absolute), its available energy drops from 304 to 264 thermal units, a loss of 13 per cent., back pressure 1lb. absolute. It is often claimed that the steam will be dried or even superheated, and that considerable gain will thereby result. In the example chosen, the steam will still contain 1·2 per cent. of moisture, so that evidently the drying action is not very great and will have no appreciable influence on the steam economy. The influence of throttling on the available energy in steam is best studied by the aid of the entropy diagram. Thus in Fig. 118 $B\ C$ and $A\ D$ are the pressure limits. Then the area $A\ B\ C\ D$ gives the available work. Now let the steam be throttled to the pressure $E\ F\ G$. Then the areas $B\ C\ F\ E$ and $F\ G\ H\ J$ must be equal, so that the area $A\ E\ G\ K$, which represents the available work in the throttled steam, must be considerably less than the area $A\ B\ C\ D$, the available work in the initial condition. Owing to this decrease in the available work the velocities of the steam will not now be correct, and this will lead to a further loss. In spite of these defects, throttle governing has shown itself very satisfactory, as we shall see later on. It has one advantage, namely, that the mean density of the steam being less, the friction loss will be less with the throttled steam.

Governing by Cutting Out Nozzles.—This method is best exemplified by the De Laval turbine. Here the number of nozzles supplying steam can be varied, within limits, to suit the load, thus maintaining a constant initial pressure and velocity. The available work in the steam is not altered. When we apply this method to a multiple-stage impulse turbine such as the Curtis or Rateau we are faced by a difficulty. To secure a correct pressure and velocity distribution throughout the turbine we ought to alter the nozzles for *all* the different stages in the same ratio. Whilst not impossible—it has been attempted in the Schultz turbine—this method introduces many difficulties, chiefly of a mechanical nature. Rateau therefore adopted the simple throttle governor. In the Curtis turbine, however, the first set of nozzles are provided with individual shut-off valves. These are

controlled by auxiliary steam cylinders with electrically-operated valves brought into action by a flyball governor on the turbine shaft. The succeeding sets of nozzles are continuously open, so that the pressure and velocity distribution will only be correct at one particular load. At light loads the pressure drop in the first set of nozzles will be excessive, and too small in the succeeding sets. The losses produced by this defective steam distribution will be greater the greater the number of stages, but, for reasons already discussed when dealing with blade angles, it is not desirable to reduce the number of stages below three or four. Fig. 31 will give an idea of the means by which the Curtis turbine is governed, the illustration showing three out of five nozzles in the first stage closed.

Varying the Time of Admission.—This is the method usually adopted with the Parsons turbine. The idea is to admit high-pressure steam in blasts or puffs, the duration of which shall depend on the load. In this way it is thought to secure the full benefits to be derived from

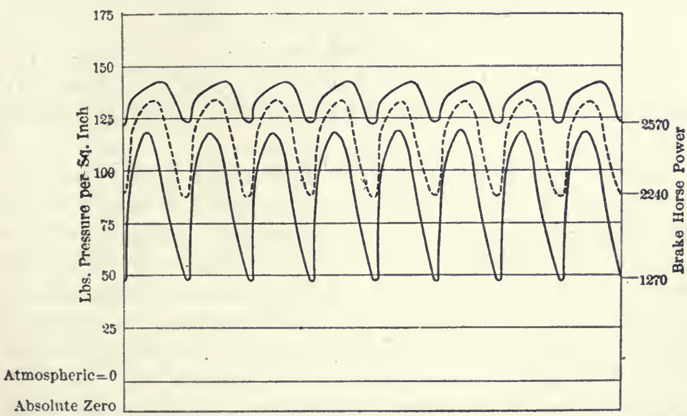


FIG. 119.—INDICATOR CARDS SHOWING INITIAL PRESSURES IN A WESTINGHOUSE-PARSONS STEAM TURBINE.

the use of high-pressure steam at all loads. As a matter of fact, this is not quite the case. In the first place, when the admission valve closes (or nearly so) there is still steam left in the turbine, and this proceeds to expand, giving a falling pressure at the high-pressure end. This is illustrated by Fig. 119, which shows the pressure variation

inside the admission valve of a 2,000 h.p. Westinghouse turbine at different loads up to about 30 per cent. overload just before the by-pass comes into action. Evidently, then, the full advantages of high-pressure steam are not attained. It should be pointed out that the effect of this pressure variation is not quite the same as that due to throttling. In throttling the fall in pressure is due to a conversion of kinetic energy—derived from the fall in pressure—into heat. In the Parsons turbine the fall in pressure is due to a removal of part of the steam to the exhaust, this steam having its kinetic energy absorbed by the moving blades and not rubbed down into heat. The reduced pressure will, however, lead to incorrect steam velocities which are a source of loss.

A glance at Fig. 119 shows that there is a certain amount of pure throttling with this type of governor. Thus at 2,570 b.h.p. the maximum pressure just inside the steam space is about 145lbs. per square inch, whilst at 1,270 b.h.p. the maximum pressure is only 120lbs. Further, the maximum pressures do not indicate the full amount of throttling actually taking place.

Some firms making the Parsons turbine are now recommending the use of a throttle governor especially where the turbine has to run in parallel with reciprocating engines.

The chief disadvantage of this method of governing lies in the fact that it gives rise to *initial condensation.* The blasts of steam occur about once in every 30 revolutions of the turbine, so that they are sufficiently distant from one another to admit of the interior of the turbine being cooled by the falling temperature between the blasts. When a fresh blast occurs the steam meets the relatively cold surfaces, and condensation must take place exactly as in a reciprocating engine. The fact that the heat thus given to the metal is returned to the steam during a falling pressure does not get rid of the loss, as will be evident from a consideraton of the entropy diagram, the heat being lost at a high temperature and recovered at a lower one just as in a reciprocating engine. Incidentally it may be pointed out that this periodic pressure variation gives rise to a periodic variation of the turning moment and end thrust on the casing which will

give rise to vibrations in the frame and be transmitted to adjoining structures.

By-pass Governors.—The area of the steam passages increases as we approach the exhaust end. Consequently the quantity of high-pressure steam which the turbine can pass will depend upon the position at which it is admitted; by admitting nearer the exhaust we obtain more power. The obvious objection to this method is that the greater the power the fewer the number of active rows of blades and the greater the number which are left running idle in a bath of dense high-pressure steam. This method is, then, obviously uneconomical of steam, and is never used except to enable the turbine to take an overload. By using this by-pass for overloads the turbine is only sufficiently large to enable it to deal (normally) with its normal full load, at which consequently it is most economical. If an overload has to be taken without a by-pass, the turbine must be of correspondingly greater capacity and its maximum economy will only occur at the overload, where it is of less benefit than at 0·75 load or full load. The Curtis turbines use no by-pass; Parsons and Rateau do. The Westinghouse turbine has its overload valve under the control of the governor, so that it automatically opens when an overload comes on. This is especially useful in traction power plants, but is seldom necessary in electric lighting or factory plants, where the change in load is known before it comes on and can be accommodated by opening a by-pass valve by hand.

In order to make a strict comparison between these different types of governing we ought to have for reference turbines of the same type except as regards the governor. The electric generator, where one is used, should have the same efficiency in all cases. In the absence of these desirable conditions we can make a very instructive comparison by taking the percentage total steam consumption at certain definite fractions of the most economical load. Fortunately, neither the power (within wide limits) or the superheat and vacuum (within limits) have any marked effect on these percentage consumptions. The following table illustrates the results of tests. The loads are stated as percentages of the most efficient load.

TABLE XIII.

Load, per cent.	125	100	75	50	25
Zoelly, 350 kw.	...	100	80·3	58·8	36·3
Rateau, 1,000 kw....	...	100	78	55·6	33·7
Rateau, 500 kw....	...	100	77·2	55	32
(U S.A.) Westinghouse, 1,250 kw. ...	130	100	78·5	57·3	36
(U.S.A.) Westinghouse, 400 kw. ...	...	100	78·5	57	...
C. A. Parsons, 1,500 kw.	...	100	78	56	33·4
C. A. Parsons, 500 kw.	...	100	78	56	33·7
Brown-Boveri, 3,000 kw.	...	100	77·6	56	33·3
Curtis, 2,000 kw.	...	100	77·2	54·2	31·4
Curtis, 600 kw.	...	100	76·5	52·4	28·8
Curtis, 500 kw.	...	100	76·2	51·7	28.1
De Laval, 200 kw....	...	100	76·2	52·3	29
De Laval, 20 kw.*	...	100	82	68	52

* Throttle governing.

The Zoelly and Rateau turbines use a simple throttle governor. The 200 kw. De Laval in the particular instance referred to above was governed by varying the number of active nozzles. The Curtis turbine was governed by varying the number of nozzles admitting steam to the first stage. The Parsons, Brown-Boveri, and Westinghouse used the blast governor. The 20 kw. De Laval was governed by a throttle valve and should be compared with the 200 kw. De Laval which was governed by cutting-out nozzles. It will be seen that the Curtis turbine and the De Laval show up best so far as the method of governing goes. It must be confessed, however, that the figures for the Curtis turbine are not altogether satisfactory. The makers of this turbine publish very few data regarding the tests made, and some of the figures have the appearance of having been smoothed so as to lie on a curve. Still there seems to be little doubt as to the relatively high efficiency of this turbine at light loads. How far this is due to the method of governing adopted it is difficult to say, as the turbine is structurally so different from the others mentioned in the table. It would be interesting to have the steam consumption at various loads deter-

mined for a Curtis turbine when using a throttle governor, and also when using the present form of governor. The Rateau turbine, using a throttle governor, comes next in (relative) light-load efficiency, and is closely followed by the C. A. Parsons turbine. The Westinghouse-Parsons turbine seems to fall below the standard of the original Parsons turbine.

The Curtis method of governing by cutting-out nozzles in the first stage is not suitable for a turbine with so many stages as the Rateau or a Parsons, so that the choice for these turbines lies between a throttle and a blast governor. The latter is more complicated in its action and it is doubtful if it is more efficient. The by-pass method of governing is obviously uneconomical of steam. The by-pass is not usually brought into action on a Westinghouse turbine until about 25 per cent. overload has been reached. According to a paper by Mr. Francis Hodgkinson there is a marked drop in the speed as the overload valve is brought into play by the governor.

Some interesting results might be obtained by fitting a turbine with governors of the different types. For instance, we could determine the steam consumption curves for a Parsons turbine with blast and throttle governors; for a Rateau or a Curtis with blast, throttle, and nozzle cut-out governors.

Fig. 120 gives the result of tests on a 10 kw. De Laval turbine, running non-condensing. The curve I., II., III., IV. gives the steam consumption at various loads when the throttle governor only is in use, the boiler pressure being 140lbs. per square inch. The curve I., V. gives the results obtained when the number of nozzles was varied to suit the load, the throttle governor being out of action. The greater economy of the latter method is evident, and where conditions permit it is evidently a valuable auxiliary to the ordinary throttle governor. Fig. 121 illustrates the results of tests on a 30 h.p. De Laval turbine with a steam pressure of 125lbs. per square inch at the governor valve and about 25·5in. of vacuum; no superheat. The four different curves illustrate clearly the increased economy of the nozzle cut-out method of governing over the ordinary throttle governing. For the ordinary small De Laval turbines it is hardly worth while putting these cut-out nozzles under the control of

the governor, so that where the load fluctuates rapidly this superior economy is not attainable.

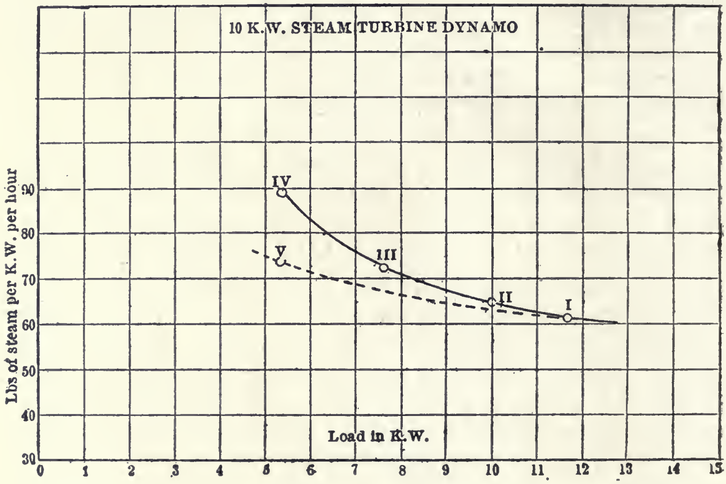


FIG. 120.—STEAM CONSUMPTION OF A DE LAVAL TURBINE UNDER DIFFERENT CONDITIONS.

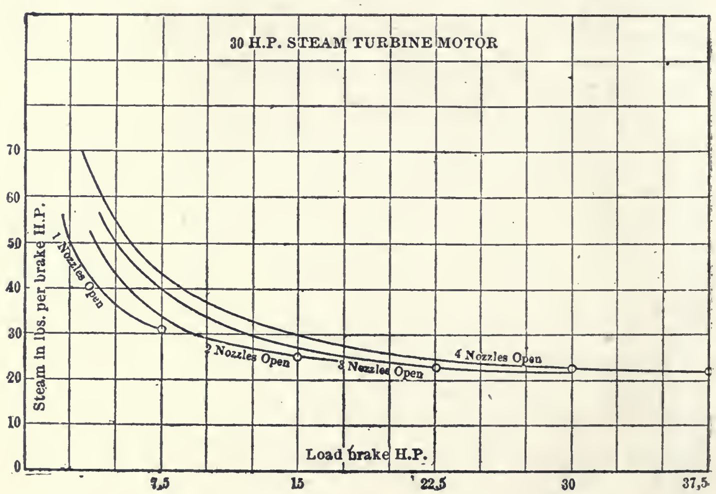


FIG. 121.—RESULTS OF TESTS ON DE LAVAL TURBINE.

By-pass for Overload.—It is important to be able to calculate the amount of overload which a certain by-pass will allow of, and vice-versa.

Suppose we have a turbine using saturated (dry) steam at 165lbs. per square inch absolute. Suppose the by-pass to lead to a point along the turbine where the pressure is normally 60lbs. absolute. Then the dryness of the steam at that point will be about 0·96, supposing that there is no initial condensation. Then the volumes of the steam, before the by-pass comes into action, at the entrance to the turbine and at the place where the by-pass enters will be 2·72 and 6·74 cub. ft. per pound respectively. Then the volume of high-pressure steam which will pass through the turbine by way of the by-pass is to the volume which would pass when the by-pass is closed, is as 6·74 to 2·72 approximately. Actually it is rather more than this, because, owing to the shorter active length of rotor and therefore steeper pressure gradient, the velocity of the steam will be somewhat greater than the normal. Assume then that the ratio is as 7·07 to 2·72, or 2·6 to 1. Then if the efficiency of the turbine under overload were as great as under normal full load, the power of the turbine would be increased in the ratio of 2·6 to 1. Owing to the reduced number of active stages and the excessive steam friction of the high-pressure blades submerged in the high-pressure steam the efficiency under overloads is much reduced. Increasing the actual number of stages in each section directly as the square of the peripheral velocities in these sections; in order to reduce them all to a comparable basis, we see that in this case the number of "equivalent" stages is reduced by from 25 to 30 per cent.* The work done in each of the remaining stages will probably be somewhat greater than under normal conditions, but this will be wholly or partially counteracted by the reduced mechanical efficiency of the turbine. Neglecting this, we see that the overload capacity is about 1·85 times the normal full load.

If we have to provide for a given overload and are not given the position of the by-pass, we reverse the order of the calculations, taking care to allow considerable margin for errors and unforeseen contingencies. For instance, suppose we have to provide for an overload

* It is sufficiently accurate, and much quicker, to take the number of "equivalent" stages as being proportional to the theoretical work done between the given pressures during the expansion.

of 50 per cent. Then we had better base our calculations on a 75 per cent. overload, seeing that this will not appreciably affect the cost or very greatly the economy of the turbine. Assume that under overload conditions the steam efficiency is reduced by 25 per cent. Then the volume of steam required at 75 per cent. overload will be about 2·33 times the full-load consumption (probably not quite so much as this). Then the steam volumes under normal full-load conditions at the main and by-pass entrances will be in the ratio of about 1 to 2·2. If the steam is supplied dry and saturated at 165lbs. absolute the normal pressure at the by-pass entrance will be about 70lbs. absolute, as this makes the steam volumes at the main and by-pass entrances as 1 to 2·2. In general, if the by-pass is calculated to enter the turbine near a change in diameter of the rotor, the by-pass entrance is situated at this change for constructional reasons.

Overload economy is not in general of very great importance, so that the use of a by-pass instead of an enlarged turbine is usually much to be preferred as being commercially more economical. In small turbines, particularly those of the Curtis type, where the turbine can be made to take a large overload without increasing its dimensions, and with the addition only of a few extra nozzles, fixed blades, and accessories, or in the case of a reaction turbine by an advantageous increase in the otherwise too small blade lengths, it may be better to do without a by-pass for overloads.

Owing to the falling off in economy when a by-pass is used, this method is not used in practice as a governing device, although it has often been proposed. Figs. 122 and 123 show two arrangements of by-pass governors. Fig. 122 shows the by-pass method applied to a turbine for which the steam is also throttled to some extent. The piston 10 is raised or depressed according as the steam pressure in the steam chest is high or low. With a high steam-chest pressure the piston rises, lifting with it the valve 7 and uncovering as it does so the ports which admit steam through the pipes 3, 4, and 5 to different points along the turbine. There is, of course, a big change in the power developed when full-pressure steam is admitted

to, say, pipe 4 instead of 3, so that intermediate powers have to be taken care of by throttling. This throttle valve is under the control of the centrifugal governor 9 or the electrical solenoid governor 12. In the device illustrated by Fig. 123, the distributing valve 7 is directly

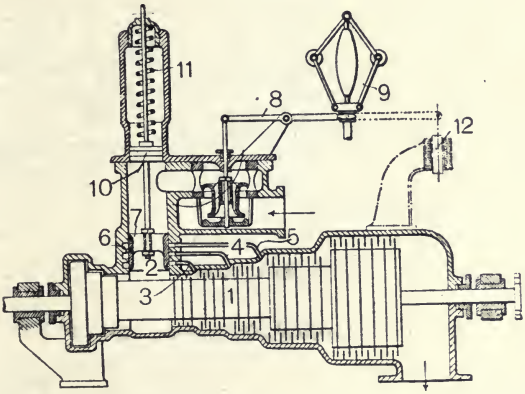


FIG. 122.—BY-PASS GOVERNOR.

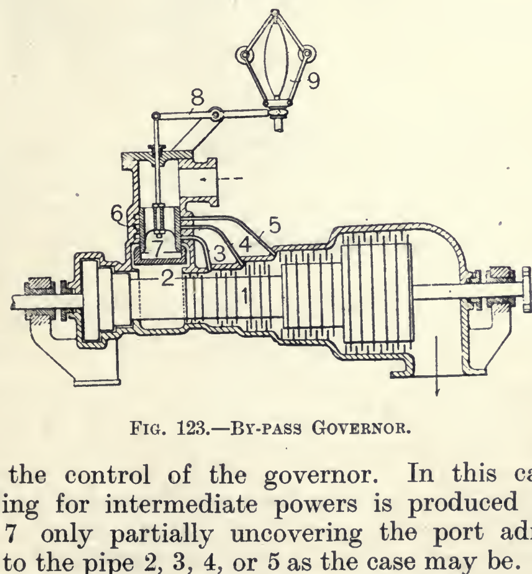


FIG. 123.—BY-PASS GOVERNOR.

under the control of the governor. In this case the throttling for intermediate powers is produced by the valve 7 only partially uncovering the port admitting steam to the pipe 2, 3, 4, or 5 as the case may be. These devices were introduced by Brown-Boveri & Co.

Fig. 124 shows the governor gear of the Zoelly turbine. The steam is throttled by the valve K which is controlled by the piston H in the cylinder G. The pressure on the piston H is produced by oil or water

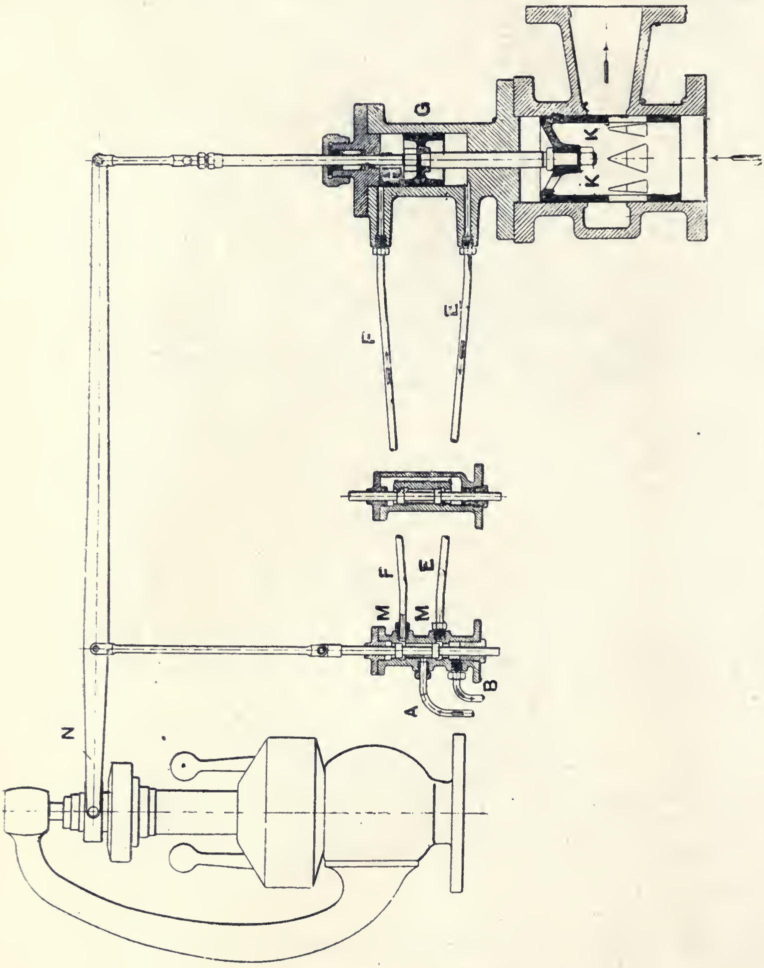


FIG. 124.—GOVERNOR GEAR OF ZOELLY TURBINE.

under pressure from a rotary pump driven off the turbine shaft by helical gearing. The valve M is a relay. Suppose the speed to rise, the centrifugal governor lifts the rod N and with it the piston valve of the relay,

thus providing a passage for the pressure liquid by way of the pipes A and F, to the upper side of the piston H, and by way of the pipes E and B from the under-side of the piston back to the suction of the pump. The pressure of the liquid depresses the piston H, thus partially closing the throttle valve and at the same time depressing the lever N carrying with it the valve of the relay, which then remains in the mid position until the speed again changes. If the speed falls due to an increase in the load, the lever N is depressed and with it the valve of the relay M. This connects the pipes F and B by way of the passage at the back of the valve shown in the detached sectional drawing of the relay, and also pipes A and E, so that the pressure liquid raises the piston H, and opening the valve K admits more steam to the turbine.

As a rule, the governor itself is similar to those used in reciprocating engine practice. In order to secure

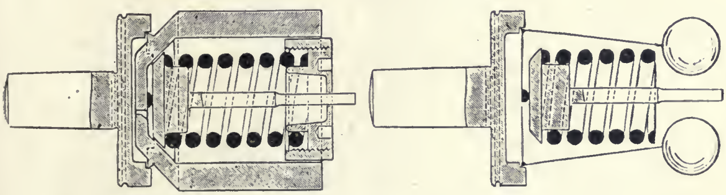


FIG. 125.—CENTRIFUGAL GOVERNOR OF DE LAVAL TURBINE.

sensitiveness, the governor weights usually turn about knife edges, and the governor spindle is sometimes carried on a ball bearing. Spring-loaded governors are almost invariably used, driven by reduction gearing of the helical type off the main shaft. The governors of the Curtis turbines are, however, mounted directly on the turbine shaft. In Fig. 32 we had illustrated the governor of a 500 kw. Curtis turbine. A somewhat similar governor is that used on the De Laval turbine and illustrated in Fig. 125. This governor is mounted on the gear wheel shaft, and consists chiefly of two leaves pivoted on a disc fixed to the shaft by means of knife edges. As the speed rises the leaves tend to separate, and in doing so compress the spring and push out the central spindle, which is connected up to the throttle valve. In some turbine governors the leaves or weights are tied

together by a spring in place of the axial spring used in the De Laval governor.

It is very common practice in steam turbine work to interpose a relay between the main governor and the

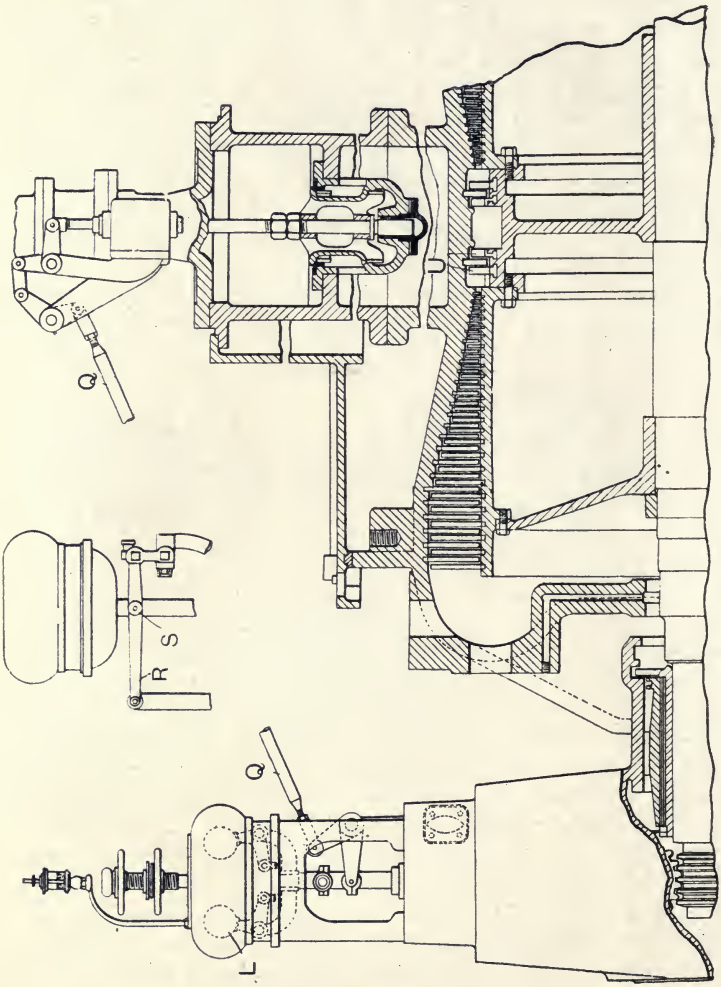


Fig. 126.—Westinghouse Steam Turbine. Governing Mechanism.

throttle valve or other steam-changing device. The main governor has thus only to move a small, light relay valve through a small distance ; the relay controlling

some powerful operating mechanism. The objections to this method are that there is necessarily a somewhat greater interval of time between a change in the turbine speed and the closing or opening of the main steam valve than with the direct-connected governor; and further-

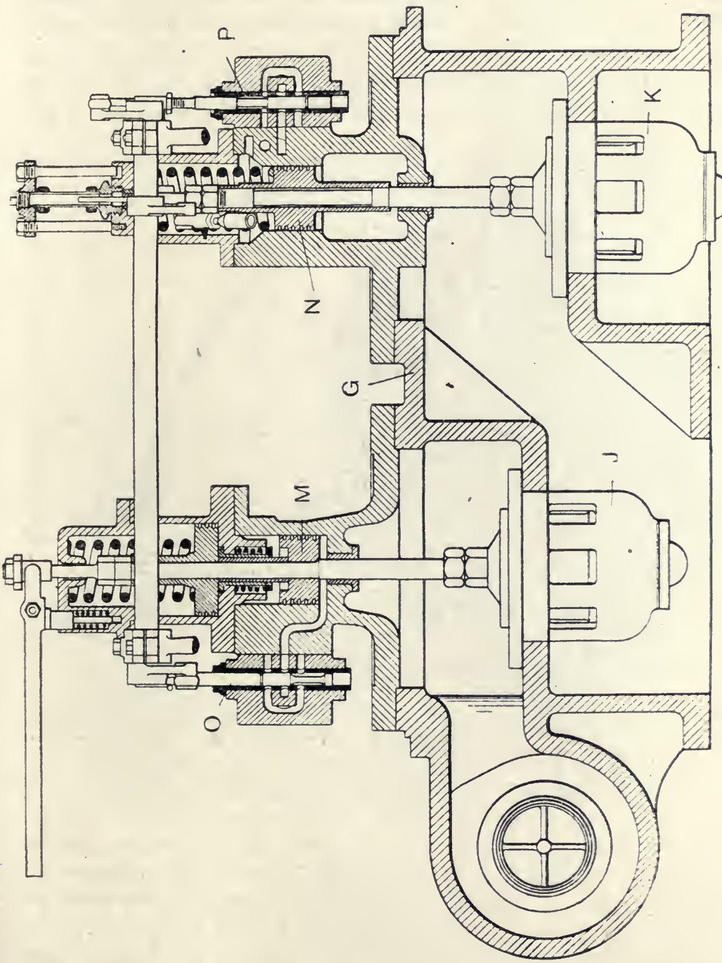


FIG. 127.—WESTINGHOUSE STEAM TURBINE. GOVERNING MECHANISM.

more the governor gear is considerably complicated. Where there are a number of small steam valves to be operated, as in the Curtis turbine, some form of relay governor is almost a necessity.

Figs. 126 and 127 illustrate the governing mechanism of a Westinghouse turbine in somewhat greater detail than was done in Chapter II. Fig. 126 shows the general arrangement of the governor levers. The rod *R* has a slight reciprocating motion—as little as 0·25in. in some cases—and, through the medium of the fulcrum *S*, causes the rod *Q* to give a reciprocating motion to the relay valves *O* and *P* (Fig. 127). The position of the fulcrum *S* is determined by the governor. At normal loads the valve *P* does not admit steam below the piston *N* which

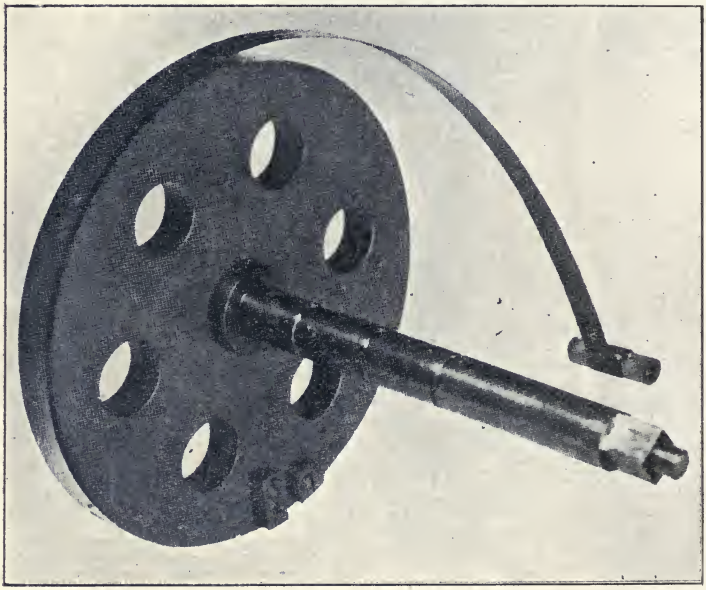


FIG. 128.—NOZZLE REGULATING VALVE.

controls the overload valve *K*, so that this valve is held to its seat. Valve *Q*, however, alternately admits steam to and exhausts it from the under-side of the piston *M* which controls the admission valve *J*. The valve *P* admits steam below the piston *N*, thus opening the overload valve *K*, when the speed falls below a predetermined limit.

Figs. 128 and 129 illustrate the governor used with the Riedler-Stumpf turbine. The disc shown in Fig. 128 is placed inside the steam chest. The strip of metal is

fixed at one end to the interior of the steam chest, and to the disc at the other end. From the steam chest pipes lead to the nozzles—see the jets of steam shown in Fig. 129—and the inlets to these pipes are closed or opened by the band of metal previously mentioned, being unwound from the disc as it is rotated by the action of the governor under the influence of a change of speed.

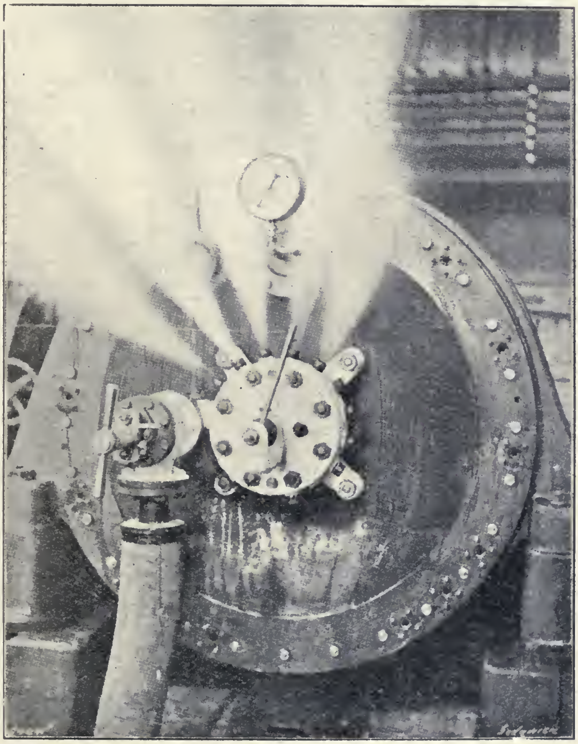


FIG. 129.—STEAM DISTRIBUTION. CHEST UNDER STEAM.

In Fig. 130 we have illustrated the general scheme of the governor gear of the Curtis turbine. The governor G is of the ordinary Hartnell type, the weights being mounted on knife edges for sensitiveness. The governor spindle acts through the medium of a bell-crank lever and a steel band B on to a short lever $A\ B$. One end B of this lever is connected directly on to the spindle of a throttle valve which controls the admission of steam to

one nozzle only. The other nozzles *N* are opened or closed by small valves, which in turn are controlled by steam pistons, the movements of which are controlled by electromagnetically-operated valves *E*. These electromagnets are under the control of a small electric drum controller *C* which is rotated by a link from the lever *A B*. The contacts in the controller are pressure—exceedingly small pressures are used—not rubbing contacts. *S* is an auxiliary controlling spring (often electrically controlled from the switchboard). The main variation of the steam supply is effected by the electrically-controlled nozzles. For intermediate powers the throttle valve *T* throttles the steam passing through one nozzle

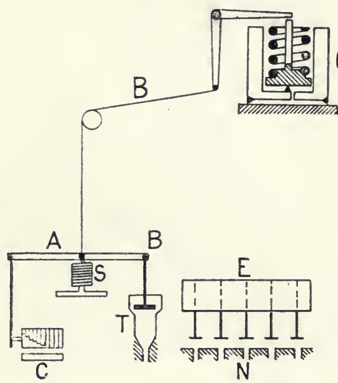


FIG. 130.—DIAGRAM SHOWING GENERAL ARRANGEMENT OF CURTIS GOVERNOR GEAR.

G, governor: Hartnell type.
B, steel connecting band.
S, auxiliary control spring.
C, Electric controller for valve magnets E.
N, Nozzles.
T, throttle valve on one nozzle only.

only. When this throttle valve *T* is full open the lever *A B* turns about *B* as a fulcrum and operates the controller C. When the valve *T* is shut, *B* is also the fulcrum; but for intermediate points *A* is the fulcrum and the valve T throttles the steam admitted to its nozzle. This governor works well in practice and is capable of exceedingly close governing.

In addition to the governor proper the ordinary 4-stage Curtis turbine has two additional governing or regulating devices. One consists of an automatic by-pass valve which, when—owing to increase of load—the pressure after the first set of nozzles rises too high,

uncovers some additional nozzles admitting steam to the second stage. In addition, one or two hand-operated valves—the valves consisting merely of plates covering nozzle entrances—are provided for the purpose of increasing the number of nozzles admitting steam to the third stage. These hand-operated valves cannot, of course, be used when the load fluctuates rapidly.

Needless to say, these different governing devices are all set to the best advantage when making a test, and hence the light load and overload economy on test is better than that commonly obtained under working conditions. This fact should be remembered when studying the economy figures at different loads as obtained under test conditions (see Table XIII.).

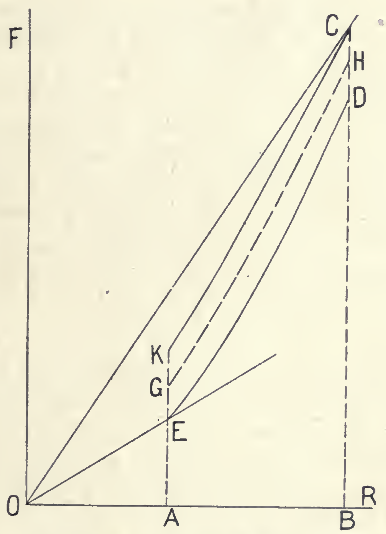


FIG. 131.—HARTNELL DIAGRAM FOR GOVERNOR.

As the governor calculations are in the main the same as for a reciprocating engine we shall not here deal with them at any great length. They are best studied by the aid of Mr. Hartnell's diagram.

Thus, referring to Fig. 131, we measure force along the vertical axis $O F$, and the radius of the governor weights along $O R$. For any given weight-radius the

controlling spring or springs exert a certain force which can easily be stated in terms of a radial force acting at the governor weights. In this way we draw in the curve $G\ H$, showing the relation between the spring-controlling force and the radius of the weights. But we have so far neglected the friction of the mechanism (including that of the linkage and valve). This friction is always such as to retard the motion of the weights. When the weights move outward (increase of speed) the pull of the spring on them is increased by the friction, so that the controlling force is now greater than that previously calculated, and may be represented by the curve $K\ C$. During a decrease in speed the weights move inward, and hence the controlling force is reduced by the friction, and may be represented by $E\ D$.

Now, at any given speed the centrifugal force acting on the weights is proportional to the radius of the circle described by them. Hence we can draw in a series of *constant-speed* lines, such as $O\ C$ and $O\ E$, showing the relation between the resultant centrifugal force of the mechanism and the radius of the weight circle. Suppose now that the speed is that of the line $O\ E$ and that the speed is falling; then at the radius $O\ A$ we see that the centrifugal force and the controlling force are the same, so that evidently $O\ A$ is the radius which the weights will assume. Similarly, if the speed is $O\ C$ and rising, the radius of the governor weights will be $O\ B$.

In calculating the centrifugal force we must take into account all revolving parts of the mechanism which are connected with the governor weights, and reduce these to an equivalent force at the weights. For instance (Fig. 132), if we have a weight W pounds connected by the lever $W\ K\ B$ turning about the pin or knife edge K to the governor ball B, and revolving at N revolutions per minute in a circle of radius R feet, the centrifugal force on this weight is

$$f = \frac{4\,\pi^2 N^2\ W\ R}{3{,}600\ g}$$

$$= 0{\cdot}000341\ N^2\ W\ R \quad . \quad . \quad . \quad \text{lbs.}$$

Then, if the governor weight B and the weight W are distant H and h from the knife edge K, measured

parallel to the governor spindle, the equivalent centrifugal force at B due to W is

$$F = \frac{f\,h}{H} = \frac{0{\cdot}000341\ N^2\,W\,R\,h}{H}.$$

Occasionally this equivalent centrifugal force may happen to be negative. Although we cannot accurately calculate the effect of friction before setting the governor to work, yet when in operation we can estimate its effect and thus obtain valuable data for future reference. The method of procedure is briefly as follows: Starting with very little load on the turbine or engine, gradually increase the load, and as the speed falls note simultaneously the speed and the position of the weights—from the position of the governor levers. In this way we obtain

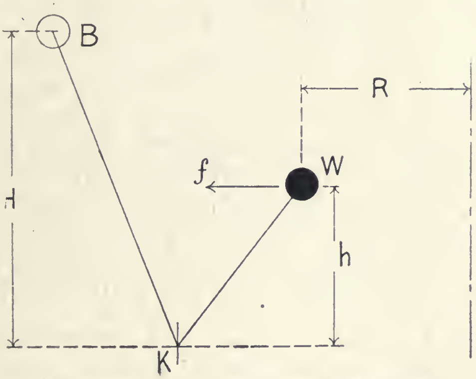


FIG. 132.—FORCES ON GOVERNOR MECHANISM.

a series of points on the curve $E\ D$, for, knowing the speed and radius, the centrifugal force—which is equal to the resultant controlling force—is at once calculable. Afterwards, gradually reduce the load, and as before note speeds and weight radii. In this way we obtain the curve $K\ C$. Both the curves $K\ C$ and $E\ D$ must have a steeper inclination than any line drawn from O and meeting them. The range of movement of the governor weights $A\ B$ is, of course, easily calculated from the maximum and minimum valve openings. Then a little consideration will show us that for this range of movement the maximum and minimum speeds are represented by $O\ C$ and $O\ E$ respectively.

In designing a governor gear it is usually best to proceed somewhat as follows : We are given the maximum and minimum speeds. We determine upon a certain scheme or arrangement of the governor mechanism. Then, knowing from the size of the steam pipe (or the relay, if used) the maximum and minimum valve openings, we can at once, from the geometry of the mechanism, calculate the range of movement $A\ B$ (Fig. 131) of the governor weights. We know from experience about what size of governor will be required, and having decided this point we can at once draw in the maximum and minimum speed lines $O\ C$ and $O\ E$. Then estimating a certain value for the friction and knowing the *relative* values of the spring-controlling force for different weight radii, we can at once draw in the curves $E\ D$ and $C\ K$, because we can take it that the friction force $K\ E$ will not be very much less than the friction $D\ C$. This determines for us the directions of the curves $E\ D$ and $K\ C$, and the actual values of the spring-controlling force are then obtained by drawing in the middle line $G\ H$. It is often more convenient to draw in the curves $K\ C$, $G\ H$, and $E\ D$—to an unknown scale—first. Then $O\ E$ and $O\ C$ give us the scale of these curves. It is, of course, inadvisable to make too small an allowance for friction, and it is usually desirable to have an auxiliary adjustable controlling spring as well as the main spring.

The above calculations refer to the permanent changes of speed. There is frequently a momentary greater change of speed than the above, whenever the load changes. This is due to the fact that a certain length of time must elapse between a change of speed and the change in the power developed by the turbine, due to the alteration of the valve.

For instance, suppose the kinetic energy stored up in the rotor of a turbine to be 3,000,000 foot-pound-second units. Suppose all load—including even mechanical friction—to be removed, and that previous to the closing of the steam valve, but after the removal of the load, 20lbs. of steam, each containing 250,000 foot-pounds of available work, enter the turbine. The (hydraulic) efficiency of the turbine is, say, 60 per cent., so that an amount of kinetic energy is imparted to the rotor equal to

$$20 \times 250{,}000 \times 0 \cdot 6$$
$$= 3{,}000{,}000 \text{ foot-pounds.}$$

Since there is no external work being done by the turbine, this is all spent in increasing the kinetic energy of the rotor. Consequently the kinetic energy will be doubled, the speed (of the rotor) will be increased by 41 per cent., and the centrifugal stresses in the rotor doubled.

The above example represents an exaggerated case, although it will serve to illustrate the importance of having a governor which is quick to follow changes of speed. In order to obtain such a governor it should be powerful —large governor weights—the governor mechanism and valve to be as light as possible.

The actions which follow the removal of the load are somewhat as follows : There is now an excess of steam admitted to the turbine which increases its speed and also that of the governor which is geared to it. This increase of speed tends to make the weights revolve in a larger circle ; but in doing so the weights have to be accelerated, so that during such acceleration the radius of the weight circle must always be less than the radius which it will assume under steady conditions at the same speed, so as to insure that the centrifugal force on the weights is in excess of the controlling (spring and friction) force ; this excess being, in fact, the accelerating force. Consequently, the speed of the turbine is higher than would be attained in steady running with the same valve opening. If the lag of the governor is serious, there will be a considerable momentary increase in speed, such as was illustrated by the numerical example. If the rotor is light, it will accelerate very rapidly, and the excessive speed attained may cause the valve to close too much, in which case the speed will then fall, and in this way oscillations of speed (hunting) may be set up. In order to check this, a dash pot is usually attached to the governor, which retards the rapidity but not the extent of the governor changes. The more sensitive the governor, the more liable it is to hunt.

Emergency Governor.—Owing to the fact that the centrifugal stresses in the turbine and generator increase as the square of the speed, an emergency governor which will completely shut off the steam supply at from 10 to 15 per cent. over speed should always be provided.

Chapter VII.

BEARINGS.

Types.—The bearings are usually of cast iron lined with white metal. A spherical seating is frequently used, and is especially desirable where the ratio of the length of the bearing to its diameter exceeds 3 or 4. The British Westinghouse and the De Laval turbines usually have spherical seats to the bearings, some of the others having a cylindrical seating. Ring oiling is occasionally used, and sight feed lubricators are used on the De Laval, but it is more usual to apply some method of so-called forced lubrication. For high speeds (say, above 2,000 revs. per minute) the Parsons firms use a special type of bearing consisting of several concentric tubes with small clearances between them. The inner tube forms the bearing proper, and the others, in conjunction with the films of oil which separate them one from another, form a semi-elastic cushion which admits of some self-adjustment and whipping on the part of the shaft. The inner bush is of gun metal. The length of the bearing depends upon the speed and weight which has to be supported, as we shall see presently; it usually has a value equal to from 2·5 diam. to 3 diam. The length of the bearings of the main shaft of a De Laval turbine is equal to from 5 diam. to 7 diam. The journal speed for a Parsons turbine varies from 30ft. to 50ft. per second. According to Mr. London, of the British Westinghouse Company (Proceedings of the Institute of Electrical Engineers, 1905, Part 173, Vol. XXXV.), 50ft. per second is the usual figure adopted in steam turbine work, and the product of the pressure on the bearing in pounds per square inch, and the journal speed in feet per second, is about 2,500 and occasionally as high as 3,000. Mr. Chilton, of the Brush Electrical Engineering Company (Institution of Electrical Engineers, February, 1904), gives the journal speeds of a 1,000 kw. turbine as 32·5ft.

and 40ft. per second, the bearing pressures being 38·5lbs. and 46·5lbs. per square inch. The maximum value of the product of the journal speed and bearing pressure is here only 1,860—a much lower figure than that given by Mr. London. The journal speed of one of the larger De Laval turbines is 87ft. per second; of a 2,000 h.p. Riedler-Stumpf turbine running at 3,000 revs. per minute, it is about 84ft. per second, and for one of the smaller Rateau turbines speeds varying from 20ft. to 40ft. per second were used.

Pressures of from 80lbs. to 90lbs. per square inch are not unusual in marine practice, the journal speed having then to be kept below 30ft. per second.

Forced Lubrication.—Except in the case of a footstep bearing, forced lubrication does not force the oil between the sliding surfaces against the pressure on the bearing. What it does is to insure a plentiful supply of oil to the bearing; after that the adhesion of the oil to the shaft will, with proper bearing proportions, insure thorough lubrication. The pressure of the oil used varies somewhat, but is seldom more than from 10lbs. to 15lbs. per square inch at the oil pump. About 0·05 gals. of oil per minute per square inch of bearing surface is usually allowed. This enables us to determine the capacity of the oil pump. The oil should always be fed to that point on the bearing where the pressure is least and no outlet provided at the portions under pressure. The place to supply the oil for a turbine bearing is, of course, the top half of the bearing. The oil grooves should always have their edges scraped or cut away, as this materially assists the lubrication.

A turbine bearing differs from a crank-shaft bearing in that the pressure on it does not vary in direction or (appreciably) in amount. When working with a full supply of oil, the alterations of pressure on a crank-shaft bearing cause the shaft to act as a suction-pump plunger, which draws in the oil to the bearing on one side whenever the pressure on that side becomes negative, thus replacing the oil previously squeezed out by the positive pressure. In the turbine reliance must be placed on the adhesion of the oil to the shaft. This is greater the greater the journal speed and the viscosity.

The factors affecting the friction of a bearing are the thickness of the oil film, the viscosity of the oil, and the journal speed. The pressure on the bearing, the temperature and quantity of the oil supply do not directly affect the friction, but affect it *indirectly* by causing some change in the thickness, viscosity, or velocity of the oil film.

It will be noticed that nothing has been said as to the influence of the materials from which the bearing and journal are formed. There are two ways in which the material may influence the friction. In the first place, the metal surfaces may not be equally smooth and consequently the inequalities in the surfaces will lead to variations in the thickness of the oil film. With a well-finished journal and bearing these variations should not vary much, and hence their influence on the friction will be small. It may be mentioned here that the oil in actual contact with the metal surfaces is stationary, so that—except when the oil supply runs short and metallic contact occurs—the friction is purely fluid in character, there being no friction (work) between oil and metal. The rate at which the metal surfaces will receive and transmit the heat due to friction will, by influencing the temperature and therefore the viscosity of the oil, affect the friction. This influence will hardly be very different for the different materials, so that we see that the kind of materials used for the bearing surfaces will have very little effect on the friction. Careful experiments by Lasche* confirm this, there being very little variation of the friction with the materials.

In Fig. 133 we have several curves showing the variation of the viscosity with temperature for different lubricating oils. It will be noticed that the viscosity decreases with an increase in the temperature, this decrease being much more rapid at low than at high temperatures. It will be noticed, too, how the different oils approach a common value for the viscosity below the temperature of boiling water. Evidently the bearing friction with different lubricants will show most variation at low-bearing temperatures, the results being pretty much alike

* "Bearings for High Speeds," by O. Lasche: "Traction and Transmission." Vol. IX.

at ordinary bearing temperatures. This is illustrated in Fig. 134, which represents the results of experiments by Lasche with different lubricants at different temperatures, but constant pressure and velocity. Lasche measured

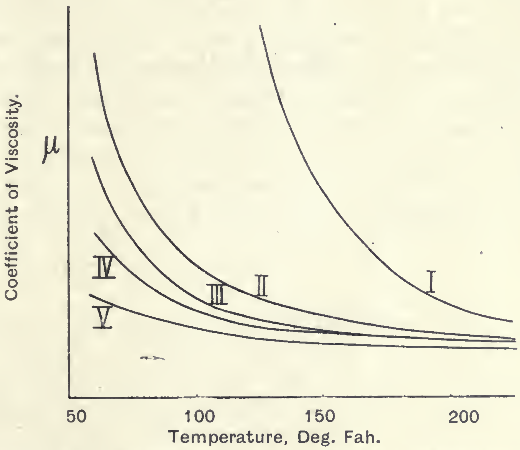


FIG. 133.—VISCOSITY OF OILS AT DIFFERENT TEMPERATURES.

I. Castor Oil.
II. Rape Seed Oil.
III. Olive Oil.
IV. Seal Oil.
V. Sperm Oil.

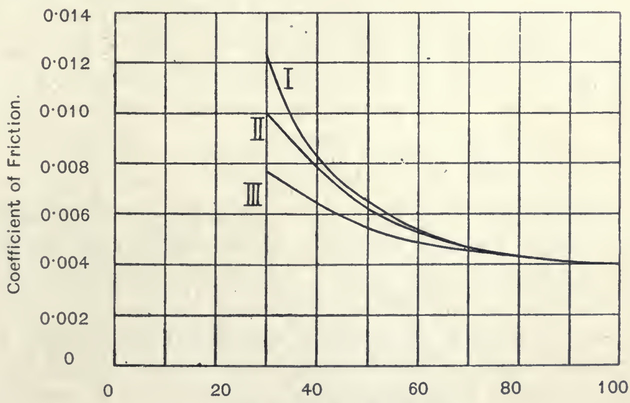


FIG. 134.—INFLUENCE OF LUBRICANT.

I. Imperial Mineral Oil.
II. Rape Seed Oil.
III. Sperm Oil.

the bearing temperature at the lowest point on the journal distant about one-third from the end of the bearing. The bearings in these experiments completely

encircled the shaft. The supply of oil was either warmed or cooled, so as to give the required bearing temperature. Full-size bearings were used.

Although we know that the friction (work) is proportional to the velocity, the area of the journal enclosed by the bearing, the viscosity, and inversely proportional to the thickness of the oil film (at the point considered), yet the conditions in a bearing make it well-nigh impossible for us to make more than an approximation of the effect of pressure, temperature, and velocity on the friction. Fortunately, we have the valuable experiments of Lasche and Tower available. Fig. 135 is from Lasche's experi-

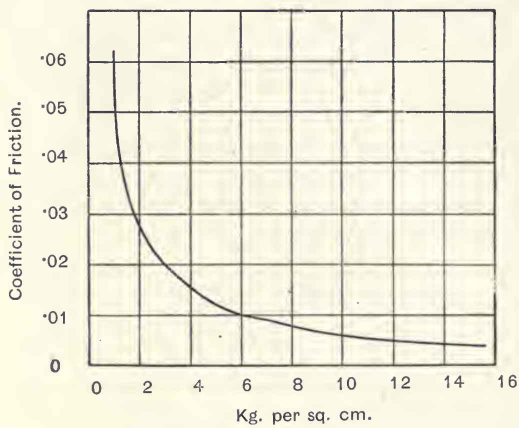


FIG. 135.—INFLUENCE OF PRESSURE.

ments when the velocity and temperature remained constant, the pressure being varied. Imperial oil, a Russian mineral oil, was used, a full (and definite) supply being maintained in all experiments. It will be seen that the coefficient of friction—obtained by dividing the total friction by the total bearing pressure—is very closely inversely proportional to the pressure intensity. With a journal completely enclosed—as this was—by the bearing the variation of the friction with the least thickness of the oil film (that in the line of action of the load) is very small except for very large changes in this thickness owing to the fact that a thinning of the film at the bottom of the bearing is accompanied by an equal

thickening at the top.* Paradoxical as it may seem at first sight, yet it will be found that the friction will *decrease* slightly as the least thickness of the oil film gets less, due to a partial failure of the oil supply, or an *increase* in the pressure or temperature.

Effect of Pressure.—Returning to the experiments of Lasche on the effect of the pressure we see that there will be a slight decrease in the friction (only notable at high pressures) due to the change in the thickness of the oil film as the pressure increases. Otherwise, the total friction will be unaltered, and hence the coefficient of friction will be very nearly inversely proportional to the pressure. This was clearly brought out in the experiments. (Fig. 135.)

Effect of Velocity.—The temperature here meant is, as previously explained, the temperature of the bearing. The heat generated by the bearing friction is dissipated by radiation and conduction from the bearing, and by being carried off in the oil. For a constant bearing temperature the first two of these channels (radiation and conduction) will carry off a constant quantity of heat, so that an increase in the friction must be accompanied by an increase in the temperature of the oil (since the oil carries away more heat). This will decrease the viscosity and least thickness of the oil film, and therefore also the friction.

Now the friction (other things being equal) will be proportional to the journal speed (velocity). Hence with a constant bearing pressure and temperature we have the increase in the friction due to the velocity, and the decrease produced by the reacting of the friction on the other conditions. The final results are represented

---

* If at any given point on the bearing the film thickness is x, the viscosity q, and the velocity v, the friction at that point will be proportional to

$$\frac{q\,v}{x}.$$

But in order to pass a given quantity of oil—neglecting that squeezed out at the ends of the bearing—v will vary inversely as x, so that the friction will be proportional to

$$f = \frac{q}{x^2}.$$

The total friction of the bearing can then be found by summing up the values of f for all points on the bearing. The viscosity q will vary as we pass round the bearing, owing to the increase in the oil temperature.

in Fig. 136, which shows Lasche's results. At high velocities the coefficient of friction becomes stationary, and even falls off.

Effect of Temperature.—The effect of an increase in temperature is to reduce the viscosity (this reduction being most marked at low temperatures) and hence also the friction and the least thickness of the oil film, which again leads to a slight decrease in the friction. Lasche's results are shown in Fig. 137 and Fig. 134, the latter showing the variation of the coefficient of friction with the lubricant used.

Effect of Clearance.—Experiments by Lasche and the results of experience show that the clearance between the journal and the bearing has a marked effect on the friction. The greater the clearance the less the friction, due, of course, to the increased average thickness of the

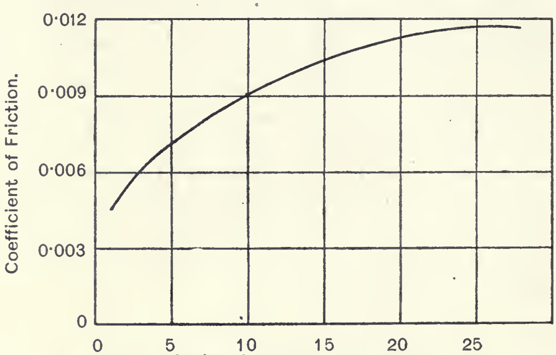


FIG. 136 —INFLUENCE OF JOURNAL SPEED.

oil film. Mr. London states that it is common practice in turbine installations to raise the top half of the bearing up about a sixteenth of an inch with very beneficial results. The top half of the bearing might, indeed, be removed altogether—except in marine turbines—and a small strap used in its place. This would greatly reduce the area of the bearing surfaces and hence the friction. The pressure on the bearing being merely that due to the weight of the turbine rotor there need be no fear of the journals lifting out of the bearings. The same thing might be done with dynamos and motors, a cover being used to keep out the dust.

Disc Oiling.—Ring oiling not being suitable for railway work, the A. E. G. * of Berlin have used a disc encircling the shaft and rotating with it. The bearing is cut in two at the middle, the disc coming between the two halves. The disc is partially immersed in an oil bath. It was found that the disc insured a plentiful supply of oil to the bearing. Such an arrangement might be useful in turbines where the journal speeds are not too high.

Ring Oiling.—Ring oiling has not been found suitable for very high speeds. Data on this subject are wanting, but a few governing principles may be pointed out. Let the weight of the ring be W and its linear speed v. It carries up to the top of the journal a weight of oil w in excess of that carried back to the oil bath. Then if F is the friction between the ring and the journal which is

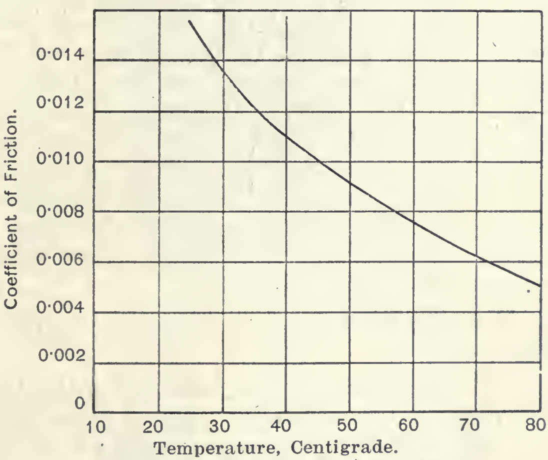


FIG. 137.—INFLUENCE OF TEMPERATURE.

causing the ring to rotate, and R is the resistance of the oil bath (depending on the depth of immersion), we must have

$$F\,v = R\,v + k\,w\,v,$$

where k is a constant, less than unity, due to the fact that the path by which the oil w is raised to the journal has a mean inclination to the vertical, and hence the work done in raising the oil is less than $w\,v$ per second.

* Allgemeine Elektricitäts-Gesellschaft.

Now
$$R = m\, v^2$$
$$F = n\, W,$$

where m and n are constants, so that we have

$$n\, W = m\, v^2 \times k\, w.$$

Evidently, then, as the velocity increases there will come a limit above which the friction F will be insufficient to overcome the increased resistance of the oil bath, and slipping of the ring accompanied by defective lubrication will result. Evidently, too, W the weight of the ring should be as large as possible relatively to w. This *requires* the use of a *large ring*, preferably *with a semi-circular cross-section*, as this reduces the ratio of the surface to which the oil will cling to the weight.

We must also consider the effect of centrifugal force. The excess of oil on the rising side of the ring will introduce an unbalanced centrifugal force, the line of action of which is obviously along a radius of the ring inclined upwards owing to the centre of gravity of this excess oil being above the centre of the ring. This force will pull the ring to one side, and will decrease the pressure of the ring on the journal, thus reducing the driving friction on the ring. Evidently, then, the centrifugal force imposes a limit on the journal speed at which ring oiling can be used. This limit is higher with a large ring than with a small one.

The De Laval turbine has wick drips provided with sight glasses.

The footstep bearing of the Curtis turbine is supplied with water (or oil; usually water) under a pressure varying from about 175lbs. per square inch for a 500 kw. machine to about 900lbs. per square inch for a 5,000 kw. machine. These pressures are sufficient to carry the dead-weight of the rotating parts. A considerable radial clearance is left round the upper bearing block to allow of the supporting fluid escaping freely. Were an oil-tight fit used, then either the pressure of the oil would not be sufficient to carry the rotor, or if it were it would raise the rotor up until the moving blades came in contact with the stationary parts. As, however, the rotor rises off the bottom bearing block the passage-way between the two blocks becomes larger, and the clearance

round the periphery of the top block permits of an escape of the lubricant with a constant reduction in pressure in the fluid until equilibrium is maintained. When oil is used for the supporting fluid a stuffing box must be placed where the shaft enters the exhaust chamber to keep out the oil. When water is used (as is more usual) no stuffing box is necessary, the water from the footstep escaping up between the shaft and a small guide bearing into the exhaust chamber.

In Fig. 30 we have a drawing of one of the earlier forms of footstep bearing for the Curtis turbine, whilst

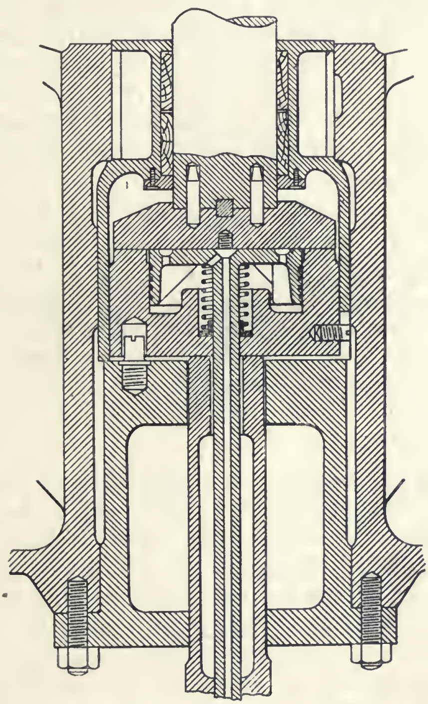


FIG. 138.—THRUST BEARING FOR CURTIS TURBINE, WITH WATER AS A LUBRICANT.

in Fig. 26 is to be seen a more modern type. Fig. 138 illustrates a Curtis footstep bearing using water as a lubricant. The clearance round the outer edge of the top bearing block will be noted, as also the lignum-vitæ bearing which is intended to keep the bulk of the

water out of the condenser. The bearing blocks themselves are of cast iron. Should the supply of water fail, the central piston of the lower block lacks support and will be pressed down by the upper bearing block, which will grind away the annular solid portion of the bottom block. When the water supply is again renewed the central piston is forced up and will carry the load so that the worn surfaces of the outer ring will not, it is hoped, prevent the normal running of the turbine.

Water Cooling.—Some large bearings are water cooled, whilst in most large turbines the lubricating oil is cooled by a coil of piping carrying cold water. One of the troubles of ring lubrication is the difficulty of sufficiently cooling the oil in the bath. This is sometimes done by a water-pipe coil. In all cases it is assisted by using a large oil bath or well, so that the time during which the oil remains in the well cooling before being used over again is increased.

The work spent in overcoming the friction of the bearings is all converted into heat, which must be removed as fast as it is generated. It is for this reason that large cooling tanks for the oil and ample radiating surfaces for the bearings must be provided. It is also very good practice to give the bearing a large seating on the pedestal in order to assist in getting rid of the heat. Water cooling of the bearing itself is very seldom necessary.

The chief function of water cooling is to so reduce the temperature of the oil that its viscosity or body is sufficiently increased to prevent the possibility of the pressure reducing the least thickness of the oil film to a point when there is a danger of metallic contact between journal and bearing. Consequently water cooling is, other things being equal, most needed where the pressures are high—as in marine work—or the temperatures are high, as may be the case when high journal speeds are used.

The problem of getting rid of the heat and the calculation of correct bearing proportions is further complicated by the close proximity of the turbine casing, although this has very little effect on the bearing at the condenser end of the turbine. According to Mr. Speakman, the

bearing temperature should not exceed from 140° Fah. to 150° Fah., although considerably higher values have been successfully used.

The chief objection to the use of high-bearing pressures is not that there is an increase in the total friction, as the friction is practically constant; but that, unless the velocity is increased, the minimum thickness of the oil film is reduced, and the danger of metallic contact between journal and bearing is correspondingly increased. If the velocity is increased with the pressure, the bearing temperature becomes dangerously high, as is illustrated in a negative way by the fact that in marine turbines with high-bearing pressures and low velocities the bearing temperatures are comparatively low.

Chapter VIII.

GYROSCOPIC ACTION: DEFLECTION OF DISCS.

Gyroscopic Action.—When a turbine-driven boat pitches or turns in a curve the combination of the ship's motion and the rotation of the turbine rotor causes the shaft to exert certain so-called gyroscopic pressures on the bearings. These gyroscopic pressures are not brought into play when the ship rolls about an axis parallel to the turbine shaft or when the motion of the ship is purely a vertical one, although in the latter case vertical pressures

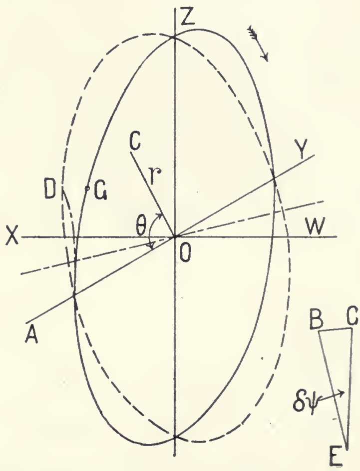


Fig. 139.—Displacement of Disc Due to its Rotation and Pitching of the Ship.

due to the change in the vertical momentum of the turbine will be called into play, but these are not properly gyroscopic.

Suppose the ship to pitch, the position of one of the turbine wheels changing from that shown by the full lines in Fig. 139 to that indicated by the dotted lines, in an interval of time δt. Then the direction of rotation

of the wheel being upwards at A we see that the point A has two motions, one in its plane of rotation and one due to the pitching of the ship, at right angles to this plane. Hence A instead of moving to G in the interval of time δt will move to D, describing a path which is *concave* to the end X of the horizontal axis. Similarly, we see that the motion of all points (on the wheel) on the same side of the vertical axis $O\ Z$ as A will describe paths which are concave to X. Also we see that all points on the wheel on the other side of the axis $O\ Z$ from A describe paths which are concave to W. Each point, then, is subjected to a centrifugal force. The resultant of all these centrifugal forces is a couple tending to twist the wheel round about the axis $O\ Z$. In addition there will be a bending moment on the shaft due to the changes in the angular momentum of the wheel about the axis $O\ Y$. This, however, is quite distinct from the gyroscopic action just described. The gyroscopic couple thus produced is balanced by two equal and opposite pressures on the two bearings whose moment is equal to that of the couple.

Let W_1 = angular velocity of wheel.

W_2 = angular velocity of the ship about the axis $A\ O\ Y$.

Consider a point C at radius r making an angle θ to the axis $O\ A$.

Then $$v = W_1 r.$$

In an interval of time δt its *direction of motion* in a vertical direction changes by an angle $\delta\psi$ from $E\ C$ to $E\ B$.*

Now $$\delta\psi = W_2\,\delta t\cos\theta$$

and if R is the radius of curvature of the path of the point, and δs is the length of this path described vertically during the interval δt, we have

$$R = \frac{\delta s}{\delta\psi}$$

but $$\delta s = v\,\delta t = W_1 r\,\delta t,$$

hence $$R = \frac{W_1 r\,\delta t}{W_2\,\delta t\cos\theta} = \frac{W_1 r}{W_2\cos\theta}$$

* The horizontal component of the motion clearly does not change its direction because of rotation about a parallel axis.

So long as W_2 remains constant the twisting moment set up will be constant, but will vary—and consequently also the gyroscopic pressures on the bearings—when the pitching motion is accelerated or retarded.

If m is the mass at the point C the "gyroscopic" centrifugal force on it is

$$= \frac{m\ v^2}{R\ g} = \frac{m\ W_1^2\ r^2\ W_2 \cos\theta}{W_1\ r\ g}$$

$$= \frac{m\ W_1\ W_2\ r \cos\theta}{g}$$

and the magnitude of its moment about the axis $O\ Z$ is

$$= \frac{m\ W_1\ W_2\ r^2 \cos^2\theta}{g}$$

and hence the total *gyroscopic twisting moment* for the whole wheel is

$$G = \frac{W}{2\,g}(k^2\ W_1\ W_2) \quad . \quad . \quad . \quad . \quad . \quad . \quad (A)$$

where W = weight of the whole rotor in pounds,
k = its radius of gyration about the shaft (not about the axis $O\ Z$) in feet.

This gyroscopic moment is balanced by a couple consisting of two equal and opposite forces P at the bearings, which are situated l feet apart. We then have

$$G = P\,l \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (B)$$

and the bending moment induced in the shaft at a distance

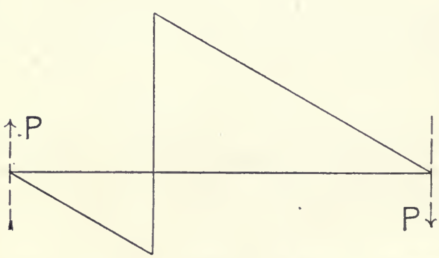


FIG. 140.—GYROSCOPIC BENDING MOMENTS ON SHAFT.

x from a bearing, but not up to the wheel, is $P\,x$ pounds-feet. The diagram of bending moments is given in Fig. 140. The maximum bending moment on the shaft occurs at the point of attachment of the wheel, and is equal to G in amount.

A similar action occurs when the ship turns in a curve.

It should be noticed that in general the " gyroscopic " bending moment is greatest when the bending moment due to inertia is least.

As pointed out by Stodola, where, instead of one wheel, we have a number of wheels uniformly spaced along the shaft between the bearings, there will be no bending moment on the shaft, only a shearing force at the bearings due to the two gyroscopic bearing pressures. Suppose there to be $m + n$ similar wheels uniformly spaced x apart. Consider a point distant $n\ x$ from one bearing and $m\ x$ from the other. If the gyroscopic pressure at the bearings due to each wheel is P then the bending moment diagram for each wheel consists of two diagrams, one for a positive moment, the other for a negative moment; both terminated at the point of application of the wheel as is shown in Fig. 140. Then at the point in question (distant $n\ x$ and $m\ x$ from the two bearings) we see that there are m positive moments each having the value $P \times (n\ x)$ and n negative moments each having the value $P \times (m\ x)$. Evidently, then, the resulting bending moment is zero.

It follows from this that the gyroscopic bending moment on a shaft of uniform diameter, unloaded, will be zero at all points; but that there will be a shearing force at all points equal in magnitude to the pressure at either bearing. In general, such a shaft would also be subjected to a bending moment induced by the pitching of the ship; such moment, however, being due to its inertia and not to gyroscopic action.

Gyroscopic bearing pressures are, of course, transmitted to the hull of the ship and, where they are of any consequence (as in lightly-built high-speed torpedo boat destroyers), it is the effect on the hull rather than the turbine itself which is generally of consequence.

To illustrate, we will consider an example. Suppose the rotor of a marine turbine to weigh 65,000lbs. and to have a radius of gyration (about the shaft) of 5ft. It rotates at 600 revs. per minute. The distance between the bearings is 15ft. The ship describes a curve under the influence of the helm of half a mile radius, at a

speed of 24 miles an hour. Then we have the following data :—

$$l = 15.$$
$$W = 65{,}000.$$
$$k^2 = 25.$$
$$W_1 = 62 \cdot 8.$$
$$W_2 = 0 \cdot 0133 \left(= \frac{24 \times 2}{60 \times 60} \right).$$

Hence the gyroscopic moment—see formula A—will be

$$G = 21{,}200 \text{ pounds-feet},$$

and hence the pressure on each of the two bearings is

$$P = 1{,}410 \text{ pounds}.$$

Again, suppose the ship to be travelling at 25 miles an hour in the opposite direction to waves travelling at 35 miles (5,280ft. to the mile) an hour, the waves being 528ft. from crest to crest, or 10 per mile. The relative velocity of the ship and the waves is 60 miles an hour, so that the ship makes 600 complete up-and-down oscillations per hour, or one in 6 secs., so that the time of an oscillation is

$$t = 6$$

If the total angle (above and below the horizontal) through which the ship pitches is

$$a = 0 \cdot 3 \ (= 17 \cdot 2^\circ),$$

then its maximum angular velocity of pitching is

$$W_2 = \frac{2 \pi a}{t} = 0 \cdot 314.$$

The other quantities being unaltered—except that the ship is now pursuing a straight course—the gyroscopic moment is

$$G = 500{,}000 \text{ lbs.-ft.}$$

The bearing pressures are therefore each

$$P = 33{,}200 \text{lbs.}$$

If the ship had been travelling with the waves instead of against them the time of an oscillation would be

$$t = 36.$$

If the amplitude a remained unaltered at $0 \cdot 3$ the value of W_2 would be $0 \cdot 052$ and the gyroscopic moment

$$G = 83{,}000 \text{ lbs.-ft.}$$

The bearing pressures would then be

$$P = 5{,}530\text{lbs.}$$

A rolling motion would not, of course, give rise to any gyroscopic moments.

We may also consider the pressures on the bearings due to the angular acceleration of the rotor. We know that if the couple produced by these pressures (Q) is

$$F = Q\,l$$

that

$$t = 2\,\pi\sqrt{\frac{W\,k^2\,a}{2\,g\,F\,2}}$$

and therefore

$$F = \frac{\pi^2\,W\,k^2\,a}{g\,t^2}$$

When the ship is meeting the waves this has the value

$$F = 4{,}150\text{ lbs.-ft.}$$

The bearing pressures are then

$$Q = 277\text{lbs.}$$

There will also be a pressure on the bearings due to the vertical oscillation of the ship as a whole which may attain a considerable value. This vertical oscillation will, of course, be much less in magnitude than the height of the waves from crest to trough. If h is the amplitude of this vertical oscillation, then the total pressure on the two bearings is

$$F = \frac{h}{2}f$$

where

$$t = 2\,\pi\sqrt{\frac{W}{g\,f}}$$

so that

$$F = \frac{h}{2}\,\frac{4\,\pi^2\,W}{g\,t^2}$$

If h = 6ft., t = 6 secs., then

$$F = 6{,}650\text{lbs.}$$

or, say a maximum of 4,000lbs. on one bearing.

These pressures are less serious in their effect than the gyroscopic pressures. The latter tend to twist the turbine and the hull of the ship. The former are of much less consequence, since they are directly balanced by

the upward pressure of the sea, and seldom set up vibrations in the hull that are of any consequence.

Waves.—In connection with gyroscopic and other bearing pressures, it may be desirable to give, very briefly, a few particulars as to waves. What we are concerned with is the worst possible set of conditions that can arise.

The theoretical shape of a wave is a trochoid; a trochoid wave, however, being seldom, if ever, exactly obtained.* Evidently, then, the minimum length of a wave is 3·1416 times its height, although it is usually from six to 20 times the height, the larger ratio being for waves over 600ft. long. The length of a wave is, of course, the distance from crest to crest. According to Sir Wm. White, the length of a wave in feet is about 5·12 times the square of its (full) period in seconds, and the speed of the wave is 5·12 times the period.

Captain Mottez, of the French Navy, reported that with long waves 26ft. high, the apparent weights of a frigate at crest and hollow had the ratio 12 to 8. That is to say, a turbine rotor weighing 65,000lbs. would, under these conditions, exert dead-weight pressures on its bearings varying between 78,000lbs. and 52,000lbs.

The height of a wave is difficult to measure accurately, but heights of 50ft. have been observed, and even 100ft. has been claimed. Waves 2,750ft. long have been observed with a period of 23 seconds, whilst a length of 700ft. with a period of 11 seconds is fairly common.

Deflection of Wheels.—A horizontal wheel carried on a vertical shaft will be bent or deflected by reason of its own weight. We may make elaborate calculations to determine this deflection, but it is very seldom of any practical value. Any deflection the wheel may have will be allowed for in the ordinary course of erection of the turbine in the shops or the power-house.

It is not quite the same with a dished wheel—that is one with a tendency to a conical shape—for when rotating the centrifugal forces acting on the wheel tend to flatten it out. This is true of horizontal and vertical wheels. The calculations for the deflection in this case

* If a cart wheel roll along the ground then any fixed point on the wheel will describe a trochoid curve in the air. Clearly there are many different trochoids depending on the distance of the joint from the centre of the wheel.

are more difficult than in the previous one, but fortunately there is no need to make them at all. The deflection can be determined and allowed for by means of a very simple experiment, as follows: If the casing is not provided with sight plugs which on removal enable us to see the wheels—no steam on—proceed as follows: Remove a portion of the casing—usually the upper half in a horizontal turbine—and run the turbine up to speed, preferably advancing by steps. The deflection of the discs can then be observed, and allowances made.

These deflections are only of consequence at high speeds—they increase as the square of the speed—and can be reduced by making the amount of dishing small. At very high speeds the wheels should not be dished at all.

For the benefit of those who may desire to investigate the matter more closely we will point out the method of procedure.

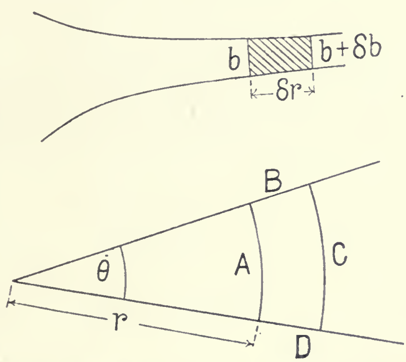


FIG. 141.—ELEMENT OF DISC ON A VERTICAL SHAFT.

Consider a small element of the wheel (Fig. 141). As we proceed outwards from the centre the thickness (b) and the bending moment (M) will usually decrease, but that does not affect the method of attacking the problem.

Apart from the shearing forces the element $A\ B\ C\ D$ is acted upon by four moments as follows:—

M_1 on the face A.

M_2 on the face C.

M_3 on faces B and D.

and δM the increase of the bending moment caused by external forces or its own weight.

These must reduce to zero when combined, so that if
M_1 gives rise to a maximum stress p,
M_2 gives rise to a maximum stress $p + \delta p$,
M_3 gives rise to a stress f,
we have, equating the moments

$$\frac{p\,r\,b^2}{6} - \frac{(p+\delta p)\,(r+\delta r)\,(b+\delta b)^2}{6} + \frac{f\,b^2\,\delta r}{6} = \delta M$$

This simplifies to

$$6\,\delta M = f\,b^2\,\delta r - d\,(p\,b^2\,r)$$

If the shearing force is F we may write this

$$6\,F = \frac{6\,d\,M}{d\,r} = f\,b^2 - \frac{d\,(p\,b^2\,r)}{d\,r}$$

If b can be simply expressed in terms of the radius r we may substitute for f and p their values in terms of the strains (see Chapter V.) and try to solve the equations thus obtained.

At the outer radius $(r = r_1)$ the value of the radial stress p_1 is known from the stresses produced by the buckets. At the inner radius $(r = r_2)$ p_2 is zero unless attached to a stiff hub.

At any radius

$$6\,M = \int f\,b^2\,dr - \int \frac{d\,(p\,b^2\,r)}{dr}\,d\,r + C.$$

The constant C is determined by making $r = r_2$ when we have

$$6\,M_2 = C - \int d\,(p\,b^2\,r) + \int f\,b^2\,dr$$
$$= C$$

If we can determine the stresses p and f or the moments which produce them, we can very easily determine the deflection to which they give rise.

If at any radius the stresses are p and f the *equivalent radial stress* is

$$p - m\,f$$

and the equivalent bending moment is

$$\frac{(p - m\,f)\,b^2\,r}{6}$$

on a circumferential arc equal to the radius.

Now we know that there is a simple relation between the loading, shearing force, bending moment, curvature, slope, and deflection as follows :—

Let R = radius of curvature,
i = slope,
u = deflection,
I = moment of inertia of cross-section,
M = bending moment,
F = shearing force,
w = loading,
E = modulus of elasticity.

Then

$$\frac{dM}{dr} = F \text{ and } \frac{dF}{dr} = w$$

also

$$\frac{du}{dr} = i \text{ and } \frac{di}{dr} = \frac{1}{R}$$

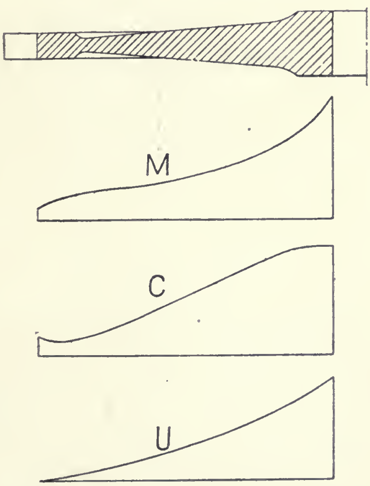


FIG. 142.
M=Curve of bending moment.
C=Curve of curvature.
U=Curve of deflection.

Evidently, then, the deflection bears the same relation to the curvature $\left(\frac{1}{R}\right)$ that the bending moment (M) does to the load (w).

But we know that

$$\frac{1}{R} = \frac{M}{EI}$$

so that we can easily determine the curvature at all points. If, now, we consider the curvature as a load, then the bending moment to which such a loading would give rise is equal to the deflection produced by the original load (w).

For instance, if in Fig. 142 the curve M is a curve of the bending moment—the equivalent bending moment in the case of a wheel or diaphragm—then we can readily determine a curve C of the curvature. Now, considering this as a loading we easily calculate the bending moment

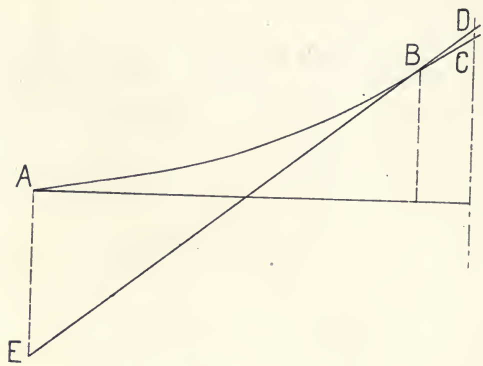


FIG. 143.—DEFLECTION OF DISCS.

curve U, which is also a curve of deflection. The question arises as to how to measure the deflection on this diagram. We know that the tangential stress at the inner radius—the maximum stress that is (being a tension at one side and a compression the other side of the disc)—is f_2, and the radial stress zero.* Then the upper side of the hole will be enlarged, and the lower side made smaller by these tangential stresses, thus giving to the disc at this point an initial slope

$$i_2 = \frac{2e}{b_2}$$

where

$$e = \frac{f_2 r_2}{E}$$

* Where the disc is attached to a cast-iron hub, this hub may be taken as fixing the slope at that point.

Then turning to our deflection diaphragm (Fig. 143), with its curve $A\ B$, draw $B\ C$ tangential to this curve $A\ B$ at B and make the angle $C\ B\ D$ equal to i_2. Produce $D\ B$ to meet the vertical through A in E. Then $A\ E$ is the deflection of the wheel at the rim. The curve $A\ B$ obviously has the same slope relative to the unstressed position that the wheel itself has ; hence the line $B\ E$ represents the unstressed position, making an angle i_2 to the stressed position at B.

When determining the deflection due to centrifugal force it is perhaps best, where the amount of dishing is considerable, to divide the centrifugal force into two components, one parallel to the disc, the other at right angles to it. These two components each produce their own deflections, which tend to balance each other.

Chapter IX.

CRITICAL VELOCITY: BALANCING.

If the speed of a turbine rotor or a generator rotor be increased there will come a certain pretty definite speed at which vibrations set in. As the speed is still further increased, these vibrations increase in magnitude, but very quickly fall off again, and at the higher speed the rotor will run quite steadily again. If the speed is increased still further, there will come another speed at which similar but less marked vibrations will set in and, on a still further increase of speed, disappear; and so on. The speeds at which these vibrations occur are known variously as the critical speeds, the whirling speeds, or the settling speeds; the latter, however, strictly refer to the speeds just after the vibrations have disappeared and the rotor has settled down to steady running. Usually, when speaking of the critical speed, the first of the series is meant.

Now, the true centre of gravity of the rotor, however well balanced it may be, is sure to receive a very slight—imperceptible it may be—displacement from the true geometrical axis of rotation. When this occurs—and in practice there is *always* such a displacement—the centre of gravity of the rotor is then describing a very small circle, and consequently the rotor is acted upon by a centrifugal force which tends to deflect the centre of gravity still more. This goes on until the stresses induced by the bending just balance the centrifugal force.

This is the critical speed of the rotor, and it exactly corresponds with the natural frequency of transverse vibration of the rotor, so that in order to determine the critical speed of a rotor all that is required is that we should determine, experimentally or otherwise, the frequency of transverse vibration for the rotor.

This relation is very easily proved.

Let–

W = weight of rotor in pounds.
u = deflection of centre of gravity of rotor from true axis of rotation in feet.
n = revolutions per *second*.
F = force in pounds weight which will bend the rotor 1ft. at its centre of gravity.

The centre of gravity is then describing a small circle of radius u, with a linear velocity

$$v = 2\,\pi\,n\,u$$

Now, the force induced in the rotor tending to prevent the bending under the action of centrifugal force is proportional to the deflection, and therefore equal to

$$F\,u$$

At the critical speed this is just equal to the centrifugal force, so that

$$\frac{W\,v^2}{g\,u} = F\,u$$

or,

$$\frac{W}{g}\,4\,\pi^2\,n^2\,u = F\,u$$

therefore

$$n = \frac{1}{2\,\pi}\sqrt{\frac{F\,g}{W}}$$

or, in revolutions per *minute* the critical speed is

$$N = \frac{60}{2\,\pi}\sqrt{\frac{F\,g}{W}}$$

Now, imagine the rotor to vibrate transversely through a distance u on each side of the axis. At the instant of passing through the axis the kinetic energy stored in the rotor is

$$\frac{W\,v^2}{2\,g} = \frac{4\,\pi^2\,n^2\,u^2\,W}{2\,g}$$

Since it has then no potential energy, this kinetic energy must be equal to the work done in bending the shaft a distance u. Therefore

$$\frac{4\,\pi^2\,n^2\,u^2\,W}{2\,g} = F\,u\,\frac{u}{2}$$

So that

$$n = \frac{1}{2\pi}\sqrt{\frac{F g}{W}}$$

n being in this case the number of complete vibrations per second. We see that it has the same value as the critical speed.

We will now proceed to discuss a few practical cases.

The simplest case is that of a uniform shaft of diameter d inches, and length between the bearings l inches. The revolutions per minute at the critical speed is given by

$$N = \frac{4{,}720{,}000\, d}{l^2}$$

When the bearings fix the direction of the shaft then we must multiply this result by 2·27 in order to arrive at the critical speed.

Let us consider the case of a uniform shaft carrying a wheel of weight W pounds. If F is the force in pounds weight necessary to deflect the shaft 1ft. at the point of attachment of the wheel, we have that the critical speed is, in revolutions per minute,

$$N = \frac{60}{2\pi}\sqrt{\frac{F g}{W}}$$

$$= 54{\cdot}2\sqrt{\frac{F}{W}}$$

And if E = modulus of elasticity in pounds per square inch (say, 29,000,000).

d = diameter of shaft in inches.

a and b = the distances of the wheel from the bearings in inches.

I = the moment of inertia of the cross-section of the shaft about a diameter.

$$= \frac{\pi}{64} d^4 = 0{\cdot}0491\, d^4$$

Then

$$F = \frac{36\, E\, I\,(a + b)}{a^2\, b^2}$$

or,

$$F = \frac{1{\cdot}77\, E\, d^4\,(a + b)}{a^2\, b^2}$$

$$= \frac{51{,}300{,}000\, d^4\,(a + b)}{a^2\, b^2}$$

In the above calculation, no account has been taken of the weight of the shaft. This may be allowed for by adding about 0·6 of the weight of the shaft to that of the wheel; not the whole of the weight being added, because the mean deflection of the shaft is not so great as that at the wheel. A more accurate method is as follows: Determine the separate critical speeds of the shaft and the wheel; let these be N_1 and N_2, then the whirling—or "critical"—speed of the combination is

$$N = \frac{N_1 N_2}{\sqrt{(N_1^2 + N_2^2)}}$$

This formula is due to Prof. Dunkerly, and is correct within a few per cent. If there are more than two critical speeds to be combined, Prof. Dunkerly gives the formula

$$N = \frac{N_1 N_2 N_3}{\sqrt{(N_1^2 N_2^2 + N_1^2 N_3^2 + N_2^2 N_3^2)}}$$

and similar formulæ for four or more speeds. These formulæ hold good when there is an intermediate bearing, or when a portion of the shaft overhangs the bearings.

Let us take as an example, a turbine wheel weighing 200lbs., carried on a shaft $1\frac{5}{16}$in. diam., and distant 12in. and 24in. from the bearings. The shaft will weigh 13·83lbs.

Then $F = 66{,}300$lbs.,
and $N_1 = 4{,}790$ revs. per minute (shaft only),
$N_2 = 986$ revs. per minute (wheel only).

Therefore $N = 967 \cdot 5$ revs. per minute (combination). If we assume that 8·5lbs. of the shaft act with the wheel we obtain a value of $N = 967 \cdot 5$lbs., which shows that our approximate method of placing about 0·6 of the weight of the shaft at the wheel, and then considering this mass alone, is reasonably correct.

In a De Laval turbine this critical speed is usually from 0·16 to 0·125 of the running speed. When starting up the critical speed should be passed as rapidly as possible, and it is desirable where practicable to start up without load for this reason.

The critical speed of a reaction turbine should always be considerably higher than the normal running speed,

as otherwise we must considerably increase the radial clearances over the blade ends in order to prevent the vibration or whipping of the rotor, which occurs when passing through the critical speed, from bringing the rotor and casing into contact. For this reason a rotor built up of broad-rimmed wheels mounted on a slender shaft is quite unsuitable. In impulse turbines this latter construction is the one usually adopted, but in this case the radial clearances can be made quite large and the critical speed can be below the running speed without serious consequences occurring, provided always that the rotor is very carefully balanced.

In a paper before the Liverpool Engineering Society,* Dunkerly gave a number of useful practical formulæ for the whirling speeds of shafts, from which the following have been selected. All dimensions are in inches, and the value of the modulus of elasticity E is taken as 29,000,000lbs. per square inch.

N = whirling speed, revs. per minute.
d = diameter of shaft in inches.

(1) *Overhung Shaft : Direction fixed at bearing.*—This case is approximated to in practice where there are two bearings close together.

l = length of overhang in inches.

$$N = 1{,}675{,}000\,\frac{d}{l^2}$$

(2) *Shaft with two bearings, overhanging one.*
c = overhang.
l = span between bearings.

Then

$$N = 1{,}000{,}000\,\frac{a\,d}{l_2^2}$$

The values of a for various ratios of c to l are given in the following table :—

$\frac{c}{l}$	1	0·75	0·5	0·33	0·25 – 0·1	very small	0·0
a	1·09	1·75	3·02	4·05	4·47	4·55	4·73

* "On the Whirling and Vibration of Shafts," by Stanley Dunkerly, M.Sc.: Liverpool Engineering Society, December, 1894.

(3) *Shaft with three bearings; one at each end and one intermediate.*

l_1 = shorter span.

l_2 = longer span.

Then

$$N = 1{,}000{,}000 \frac{a\,d}{l_2^2}$$

The values of a are given in the following table :—

$\frac{l_1}{l_2}$	1 – 0·75	0·75 – 0·5	0·33	0·25	0·2 – 0·15	0·12 - 0·1	very small
a	4·73	5·31	5·9	6·23	6·38	6·79	7·28

The preceding formulæ assume a solid shaft. If the shaft is hollow, d_1 and d_2 being the external and internal diameters, then for d in the formula

$$N = 1{,}000{,}000 \frac{a\,d}{l^2}$$

write

$$\sqrt{(d_1^2 + d_2^2)}$$

Where, as is frequently the case, the turbine and generator are in line, but only connected by a flexible coupling, the critical speeds of each must be determined separately, as they will not re-act on one another to any great extent, provided, of course, that the coupling is really flexible.

At the beginning of this section we gave a formula for the critical speed of a shaft carrying a single weight. In many cases, however, this weight is distributed fairly evenly over some portion of the shaft. In turbo-generators, the rotor frequently approximates to such a case, although owing to the longitudinal stiffness of the field or armature itself apart from the shaft which carries it, mathematical calculations as to the critical speeds of such combinations are apt to give results considerably wide of the mark.

Still, it may be interesting to give a table showing the factor k, by which the critical speed determined for a concentrated load (of the same magnitude) placed midway between the bearings, should be multiplied in

order to give the theoretical critical speed. If x is the percentage length of the shaft, midway between the bearings over which the field or armature is distributed, then the factor k is given in the following table :—

x	20	30	40	50	60	70	80	90
k	1·02	1·03	1·05	1·08	1·13	1·18	1·22	1·32

For instance, if the shaft were 10ft. between bearings, and the rotating field covered the middle 6ft., the critical speed of the field would be 13 per cent. greater than if the load were concentrated at the centre. In practice it would probably be considerably more than this, owing to the longitudinal stiffness of the field system.

General Case.—The turbine shaft is frequently of varying diameter, and as a rule will carry several wheels. Fortunately, we can modify Dunkerly's formula for the combination of several critical speeds, and use the new formula as a basis for a simple graphical treatment by which practically any case can be solved.

Unless the shaft is of uniform diameter we had better divide its weight into several portions, sufficiently small, that we may consider these weights as concentrated loads acting at the middle points of their respective territories. Where there are a fair number of wheels on the shaft, as in impulse turbines, it will greatly simplify matters to arrange that these territories are so chosen that their respective weights act at the same points on the shaft as the weights of the wheels. This will reduce the number of loads to be considered. Then, under these conditions, we have a weightless shaft carrying loads $w_1, w_2, w_3 \ldots.$, giving rise *individually* to deflections $u_1, u_2, u_3 \ldots.$ *at their points of application.* $N_1, N_2, N_3 \ldots.$ are the separate whirling or critical speeds of these loads.

Let $u = u_1 + u_2 + u_3 + \quad . \quad . \quad . \quad . \quad .$

Then we know that

$$N_1 = 54{\cdot}2\sqrt{\frac{1}{u_1}} = \frac{54{\cdot}2}{\sqrt{u_1}} \quad . \quad . \quad . \quad \&c.$$

(1) Two loads only :—

$$N = \frac{(54{\cdot}2)^2 \sqrt{\dfrac{1}{u_1 u_2}}}{54{\cdot}2 \sqrt{\left(\dfrac{1}{u_1} + \dfrac{1}{u_2}\right)}}$$

$$= 54{\cdot}2 \sqrt{\frac{1}{u_1 + u_2}}$$

$$= \frac{54{\cdot}2}{\sqrt{u}}$$

(2) Three loads :—

$$N = \frac{(54{\cdot}2)^3 \sqrt{\dfrac{1}{u_1 u_2 u_3}}}{(54{\cdot}2)^2 \sqrt{\left(\dfrac{1}{u_1 u_2} + \dfrac{1}{u_1 u_3} + \dfrac{1}{u_2 u_3}\right)}}$$

$$= 54{\cdot}2 \sqrt{\frac{1}{u_1 + u_2 + u_3}}$$

$$= \frac{54{\cdot}2}{\sqrt{u}}$$

Evidently, then, we may write as a general formula—

$$N = \frac{54{\cdot}2}{\sqrt{u}}$$

where u is the sum of the separate deflections due to the individual loads alone, at their points of application. This formula holds good whether or not there is an intermediate bearing, or if the shaft overhangs the bearings. If the shaft is of uniform diameter, we can readily calculate u_1, u_2, &c., or we can calculate N_1, N_2, &c., directly and combine them by means of Prof. Dunkerly's formula.

It has previously been shown that a very simple relation holds between the bending moment curve produced by a load and the deflection curve produced by the same load, the deflection curve being in fact the bending moment curve for the "curvature" loading. By adopting this method we can determine u_1, u_2, u_3, &c. We will illustrate this method as applied firstly to the case of a load between bearings and, secondly, to the case of a load on the overhanging portion of the shaft.

In Fig. 144 A and B are the bearings, D is the line of action of the load w_1. The bending moment curve

M produced by this load is triangular, as sketched, its height (at the point of application of the load) being equal to the product of the load, and its distances from the two bearings divided by the span between the bearings.

Construct the curve C_1, whose height C_1 at any point bears the relation to the height M of the bending moment curve at that point, given by the formula

$$C = \frac{1}{R} = \frac{M}{EI}$$

The moment of inertia (I) of the cross-section of the shaft (taken about a diameter..) will, of course, vary according to the diameter. This curve C_1 is the "curva-

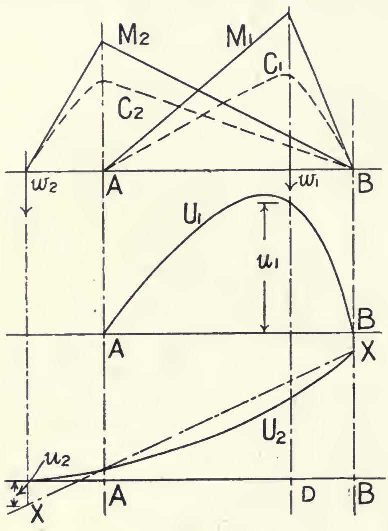


Fig. 144.—Deflection of Rotors.
M—Bending moment curve.
C—Curvature curve.
U—Deflection curve.

ture" load curve. Obtain the curve U_1 of bending moments produced by this load. Then the height of this curve at the point of application of the original load gives us the deflection u_1.

It is convenient to adopt a slightly different procedure where the load w_2 overhangs one bearing. We first obtain the bending moment curve M_2 (Fig. 144), and from it the "curvature" loading curve C_2. Then

obtain a bending moment curve for this loading in the following manner: Neglect altogether the pressures on the supports due to this curvature loading, because the change in slope of the shaft (and hence also the deflection) is determined only by the shape of the curve C_2, and to introduce a bearing pressure at A would be to introduce a discontinuity in the slope curve which a glance at the curve C_2 will show should not be there. To obtain the bending moment curve, then, take moments about the far bearing B and thus obtain the curve U_2. Now we know that U is zero at both bearings; hence draw in the base line $X X$ through the curve at the bearings, and the vertical distance between the curve U_2 and the base $X X$ at the point of application of the load w_2 gives us the deflection u_2. The quickest way to calculate a bending moment curve U from a loading curve C is to divide the area under the curve C into parallel strips and take the area of each as acting through its own centre of gravity (usually the middle of the strip).

This converts the continuous load into a number of concentrated loads, which greatly simplifies the calculations. We may, of course, employ purely graphical methods. It will be noticed that in the case of a load between the bearings the deflection at the point of application of the load (which is what we want) is not usually the maximum deflection. Although we have made use of the term shaft there is no need for the presence of a shaft, as it is usually understood. The method of procedure applies equally well where the rotor consists of a hollow drum with short shafts pressed into its ends, the drum in this case acting as a shaft.

Higher Critical Speeds.—The critical speeds so far considered have been those corresponding to a bending of the shaft, pretty much like the curve of a skipping rope. The shaft may, however, bend in other fashions, somewhat as in Fig. 145. Here we see that the deflection is zero at one or more intermediate points for the higher critical speeds. The exact critical speeds corresponding to such modes of vibration can be calculated, but in view of the awkwardness of the conditions it is sufficient in most cases to calculate the first, or lowest, critical speed and estimate the others from it by a simple approximation.

We know that the deflection of a shaft is proportional to the cube of its length, and that the critical speed is inversely proportional to the square root of the deflection, so that the critical speed must be inversely proportional to the 1·5th power of the length. Now, very roughly,

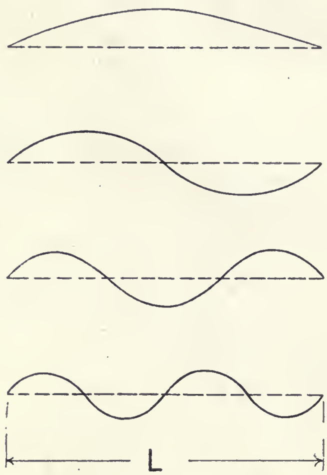


FIG. 145.—HIGHER CRITICAL SPEEDS, SHOWING MODES OF VIBRATION.

we may take the equivalent lengths of the shaft under the different modes of vibration to be respectively

$$L, \frac{L}{2}, \frac{L}{3}, \frac{L}{4}, \frac{L}{5}, \text{\&c.} \quad . \quad . \quad .$$

Consequently, the critical speeds are in the ratios—

1, 2·7, 5·2, 8, 11·2, &c.

Inasmuch as the lengths of the individual loops for any one mode of vibration are not all quite alike, it follows that the above ratios are somewhat too high. This latter clause is only applicable to the case of an unloaded shaft because the relative lengths of the loops of a loaded shaft depend on the disposition of the loads.

In the following table are given the results of some of Stodola's experiments on the critical speeds of shafts and the critical speeds as calculated by our approximation, the first critical speed of Stodola's experiments being taken as correct.

N = actual revolutions per minute.

n = calculated revolutions per minute.

N	500	1,300	2,800	7,000 (?)	Uniform shaft carrying
n	500	1,400	2,600	4,000	20 discs.
N	2,700	4,800	12,000	...	Uniform shaft.
n	2,700	7,600	14,000	...	——
N	3,200	8,200	17,000	...	Uniform shaft.
n	3,200	9,000	16,600	...	——

The close agreement between our approximation and the actual results is evident.

Balancing.—It is of the greatest practical importance that the rotor should be well balanced. To assist in securing a good balance, the rotor should be *accurately* machined all over, and the materials used should be homogeneous; this latter being more easily accomplished if forged instead of cast steel is used. The rotor must, of course, be symmetrical. It will, however, be necessary to experimentally balance the rotor. Where the rotor consists of a central drum or shaft carrying several independent wheels these should be balanced separately. It is not sufficient in balancing a rotor that its centre of gravity should be brought into the axis of rotation; we must secure in addition that the rotating masses have no unbalanced couple. For instance, the rotating mass composed of the short shaft and two pedals of a bicycle has its centre of gravity in the axis, but it is not balanced. When rotated the centrifugal forces of the two pedals are equal, but opposite in direction, thus forming a couple which owing to the rotation varies in direction continuously. The effect of this lack of a complete balance is readily observed by turning the bicycle upside down, so that it rests on the saddle and handle-bars, and rotating the back wheel. Violent vibrations will be communicated to the frame.

Experimental Balancing.—In practice, it is necessary to balance the rotor experimentally. This can be done by holding a piece of chalk close up to the shaft whilst

running. The side of the shaft which pulls out will receive a slight mark. According to Mr. London, the chalk mark is usually from 50° to 90° ahead of the out-of-balance mass, and at very high speeds is opposite to it. If the rotor is run in both directions by a motor, the out-of-balance mass will then lie midway—in angular position—between the chalk marks. Each of the two out-of-balance masses which form the couple must have its own individual balance mass.

There are several methods of obtaining a statical balance, which, however, as pointed out, does not, except in the case of an individual wheel, insure a complete balance. The most usual method of balancing (statically) a complete rotor is as follows: Support the two ends of the shaft on parallel ways, so as to make its axis horizontal. It will come to rest with the heavy side down. Balance weights having their centre of gravity in the vertical plane through the axis, and above the shaft, are then added until the rotor will come to rest in any position. A somewhat similar method is adopted for balancing individual wheels. In this case, however, it is generally more convenient to mount them (singly) on an arbor preferably with knife edges in its axis. When balanced, the wheel should not turn the arbor, whatever its position on the arbor may be, two positions at right angles being sufficient to completely test the balance. For convenience, in dealing with wheels intended to fit shafts of all diameters, one standard arbor, with a number of bushes to suit the different hole diameters, will be found most suitable. The bushes should make a tight sliding fit with both arbor and wheel, thus allowing of relative motion when desired, as for turning the wheel through 90° in the balancing process.

For convenience in balancing, the rotor or wheel should be provided with a number of small, equally-spaced pockets or holes near the circumference, in which to insert balance weights. The statical balancing of the rotor of a Parsons turbine takes from one to six days. The balancing of the generator field is quite as difficult; and unlike the turbine rotor, the field may come out of balance when running, owing to the shifting of some of the wiring.

Chapter X.

CONDENSERS.

Condensers.—In the next chapter we shall point out how greatly the vacuum affects the steam consumption of a turbine. At present it is sufficient to say that the gain in economy per extra inch of vacuum varies from about 2 per cent. at 16in. of vacuum to from 5 to 6 per cent. at 29in. of vacuum. A non-condensing turbine is not, generally speaking, so economical as a non-condensing reciprocating engine.

It would appear from the above remarks that a high vacuum is very desirable in turbine condensers, and this is undoubtedly true in many cases ; but we shall see that, contrary to the very emphatically expressed opinion of most turbine engineers, a great many turbines ought to be run non-condensing if the highest commercial economy is desired.

There are five types of condenser, as follows :—

(1) Surface condenser (ordinary).
(2) Barometric injection condenser.
(3) Non-barometric injection condenser.
(4) Ejector condenser.
(5) Evaporative surface condenser.

The main features—and we are not here concerned with the details—of the different types are too well known to need explanation. The evaporative condenser is very seldom used, as it is expensive and not satisfactory. Surface condensers are most commonly used in connection with steam turbines, but it is highly probable that the barometric condenser will come much more into favour. It has the following advantages over a surface condenser :

(1) Smaller first cost.
(2) Fewer joints to keep tight.
(3) Uses less water.
(4) Occupies very little floor space.

(5) Never requires more than two pumps (dry-air pump and circulating-water pump).

(6) Can use dirty condensing water without loss of efficiency, since there are no tubes to become coated with deposit.

The relative quantities of water used by barometric and surface condensers depend on the conditions, but are always in favour of the barometric condenser. Where the inlet temperature of the condensing water is not very different from that of the exhaust steam (as with very high vacua and hot condensing water) the surface condenser uses from two to three times as much as the barometric, whereas under more favourable conditions of water supply the difference will only be from 10 to 20 per cent.

This point is of special importance where cooling towers or ponds for the water have to be employed on account of the difficulty of getting a plentiful supply of water. The cost of a cooling tower plant is usually as large as the entire cost of the condenser, pumps, and piping, and is from three to five times this amount for a cooling pond, so that evidently a barometric condenser has an immense advantage over a surface condenser in such cases. The saving in floor space is most marked when used in connection with a vertical turbine, as it is common practice to instal the condenser directly underneath horizontal turbines. Even here, however, the barometric condenser has the advantage, as the size and cost of the basement will be much reduced.

Where dirty condensing water must be used the boiler feed cannot be taken from the discharge of a barometric condenser, which in most cases means an appreciable loss of heat.

A barometric condenser requires a large air pump on account of the air brought in by the condensing water, but this is usually offset by the decrease in the size of the circulating pump, which usually consumes about two-thirds of the total power supplied to the condenser auxiliaries. All things considered, it is to be expected that barometric condensers will largely replace surface condensers.

Counter-current and Parallel Flow Condensers.—If the general direction of the motion of the steam and the condensing water is the same the condenser is said to be of the parallel-flow type, and, conversely, if the directions of flow are opposite the condenser is of the counter-current type. The latter are usually considered the better type, because the temperature of the condensing water can be raised higher, and therefore a less quantity of water used, than with the parallel-flow type. When applied to surface condensers there are, however, three objections to this type. In the first place the condensing water is at its highest temperature when in contact with the hot steam (with the walls of the condenser tubes between them, of course), and at its lowest temperature when in contact with the comparatively cold hot-well discharge. Hence the *difference in temperature* between the water in the tubes, and the steam and water in the condenser proper is less than in a condenser of the parallel-flow type, and a larger condenser will therefore be necessary. Again, since the hot-well discharge is last in contact with the cold (inlet) condensing water its temperature will be lowered more than is desirable. Also, since most of the steam is condensed where the water is coldest it will first have to traverse the maze of tubes, the frictional resistance of which will reduce the available vacuum at the turbine. All things considered, it seems probable that the parallel-flow surface condenser is better than the counter-current type in most cases.

With regard to the increased cooling of the condenser discharge with the counter-current condenser, this is in some senses an advantage, especially at high vacua. At each stroke of the air pump there is temporary drop in the condenser pressure, which if the water be at or near the boiling point for that pressure, will cause a considerable evaporation to take place and thus increase the work required from the air pump.

So marked is this action that some firms—notably Allen and Parsons—have placed an internal weir or (its equivalent) in front of the inlet to the discharge pipe so that the bottom tubes of the condenser are always submerged. The circulation of the cold condensing water thus cools the condenser discharge and reduces the evaporation due to the pulsating action of the air pump.

This pulsating action is considerably reduced, where multiple pumps are used or where, as with a dry-air pump, the pump can be run at a high rotative speed. An air pump having a continuous-suction action such as a centrifugal pump, would, if practicable, be an improvement on the present types.

The above objections do not hold for a barometric condenser, so that the counter-current type is preferable in this case. In a counter-current barometric condenser the water enters as near the top of the barometric head as possible and the steam as low down as convenient.

Cooling Towers and Cooling Ponds.—According to a paper by Mr. W. H. Roy* cooling towers occupy a ground area of about 1 sq. ft. per h.p., or, say, 1·5 sq. ft. per kilowatt, whilst ponds require about 40 square feet per horse-power (60 sq. ft. per kilowatt). The cost of a cooling tower with foundations and a concrete tank 8ft. or 9ft. deep he gives at 16s. per horse-power (24s. per kilowatt), and that of a pond as varying from £2. 10s. to £4 per horse-power. These figures are based on 20lbs. of steam per horse-power.

In a cooling tower the water enters at the top, where the air for cooling the water is hottest and contains most moisture, and leaves at the bottom, where the air is driest and coldest. This is not as it should be. It is economical of air, but that is a very secondary consideration. The tower should be so arranged that at all points the difference in temperature between the water and air should be a maximum, and the humidity of the air a minimum. To secure this, the tower should preferably be in sections divided by horizontal partitions in each of which the air and water should (in general) flow in the same direction. By these means a much smaller tower is required, whilst the power required by the fan will at least—for a given horse-power of the main engines—not be greater than in the present types of cooling tower. The author has recently patented a tower of this type.

Design of Surface Condensers.—It is not proposed here to enter minutely into the problems of design, but rather

* "Condensing Plant for High Vacuum:" Transactions of the Manchester Association of Engineers, 1903.

to point out the guiding principles which should underlie the design.

Condensers cannot be designed with the same certainty of results as a dynamo because of our insufficient knowledge of all the conditions, and of the properties of the materials involved. A great deal depends on the condensing water, for clearly if the water can be drawn from the comparatively cool sea or from a cold lake or river, much better results can be obtained from a given condenser than can be if the condensing water is as hot as it frequently is when drawn from a canal or slowly-flowing stream in a crowded manufacturing district. Then again, the presence of even small quantities of air in the condensed steam will greatly retard condensation. It is partly for this reason that the stuffing boxes of a turbine are commonly provided with a water or steam seal.

The heat abstracted from condensed steam has first to be got into the metal of the condenser tubes, then carried across the metal to the water surface, and finally got into the water. The first and last of these are the only serious problems. This transfer of heat is greatly assisted by anything which disturbs the steady motion of the water or the steam and sweeps the condenser tubes clean of the bulk of the condensed steam. The mere fact that heat is being transmitted to the water seems to assist in producing the desired results ; still, in order to make sure that the motion of the water is sinuous and not in steady stream lines the velocity of the water ought always to exceed the critical velocity at which sinuous motion first appears. The value of this critical velocity in feet per second for a tube of internal diameter D inches is about

$$V = \frac{K}{D} \text{ feet per second.}$$

where K has the following values for various temperatures of the condensing water at the inlet.

Temp. Fah.	32	41	50	59	68	77
K.	0·47	0·39	0·34	0·29	0·26	0·23

Thus, if D is 0·375in. and the inlet temperature of the water 41° Fah., the velocity of the water ought to exceed 1·04ft. per second. In vertical condensers the water should preferably flow downwards, as this assists in disturbing the steady flow of the water.

Dr. Stanton* gives the following formulæ for surface condensers :—

L = length of tubes between inlet and outlet of water, in feet.
d = diameter of tube in feet.
v = velocity of water in feet per second.
T = temperature of the inner surface of tube in degrees Fahrenheit.
t_1 = inlet temperature of the water.
t_2 = outlet temperature of water.
t_m = mean temperature of water.

Then for smooth brass tubes

$$K L = \frac{d\,(930\,v\,\mathrm{d})^{0\cdot14}}{1+0\cdot002\,(T+t_m)} \log\left(\frac{T-t_1}{T-t_2}\right)$$

where $K = 0\cdot0105$.

The temperature of the tube (inner surface) was found to be very different from that of the steam, the two temperatures being connected by a linear law such as

Steam temperature $S = 0\cdot782\,T + c$,

where c is a constant depending on the efficiency and speed of the air pump. At low pressures (in the condenser) the difference between S and T sometimes reached 30° in Stanton's experiments. It is probably less than this in turbine work, owing to the somewhat smaller quantity of air finding its way into the condenser.

If n = number of tubes,
s = total cooling surface,
Q = cubic feet of water used per second,

then
$$s = n\,\pi d\,l$$
$$Q = \frac{n\,v\,\pi d^2}{4}$$

and hence
$$v = \frac{4\,Q\,l}{d\,s}.$$

Inserting this in Stanton's first equation, he obtained

$$K L = \left(\frac{3{,}720\,Q\,L}{s}\right)^{0\cdot14} \frac{d}{1+0\cdot002\,(T+t_m)} \log\frac{T-t_1}{T-t_2}.$$

These two equations will be found of considerable value in studying condenser design. In order to illustrate the use of the formulæ we will consider a few examples.

* "Surface Condensers," by T. Stanton, D.Sc. Proceedings of Institution of Civil Engineers. Vol. CXXXVI. 1898-99. Part II.

Example I.—Vacuum constant.

$T = 90 = \text{constant},$

$t_1 = 40 = \text{constant},$

t_2 variable.

Fig. 146 shows how the length (Curve I.) of the tubes (the total length of the water path), their surface (Curve II.), and their number (Curve III.), vary when the outlet temperature varies. It should be noted that the quantity of water used is proportional to the number of tubes, so that Curve III. is also a curve showing the quantity of

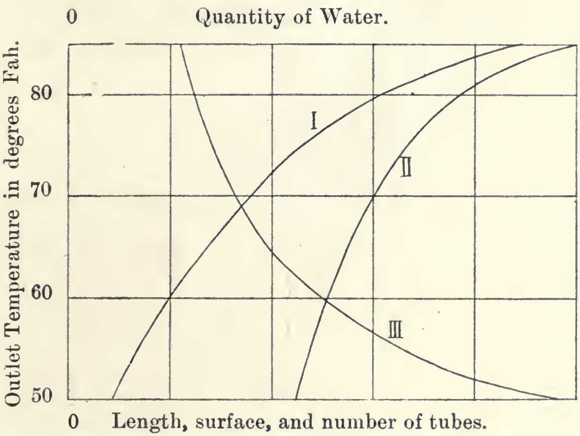


FIG. 146.—EFFECT OF OUTLET TEMPERATURE OF CONDENSING WATER ON CONDENSER PROPORTIONS.

I. = Length ; II. = Surface ; III. = Number and Water.

condensing water. It will be noticed how greatly the length of the tubes has to be increased in order to add the last few degrees to the outlet temperature of the water. It will be noticed, too, that the higher the outlet temperature the greater must be the tube surface, and hence the more costly the condenser and the more powerful the circulating pump. Clearly then, where condensing water is plentiful we may take it as a general principle that the condenser should be short. It must always be remembered that least cost and not high thermal efficiency is desired. The method of determining the best length for the condenser is based on certain general principles which will be discussed in connection with the determination of the best vacuum.

Example II.—Vacuum variable.

$t_1 = 40 =$ constant,

$t_2 = 76 =$ constant,

$T =$ steam temperature—20.

According to Stanton the tube temperature given above will be unduly favourable at high vacua. Fig. 147 illustrates how the tube length (Curve IV.), surface (Curve V.), number of tubes and quantity of water (Curve VI.) vary when the vacuum changes ; other things remaining unaltered. We see that the number of tubes and the water used hardly change at all, whereas the length and surface of the tubes goes up rapidly as the vacuum increases. Thus under the above conditions the tube

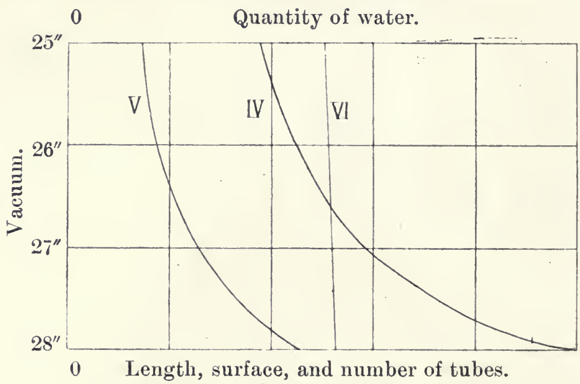


FIG. 147.—EFFECT OF VACUUM ON CONDENSER PROPORTIONS.
IV. = length ; V.= surface ; VI.= number and water.

surface necessary to obtain a vacuum 28in. is three times that required for a 26in. vacuum. Evidently, then, a high vacuum is an additional argument for the use of short condensers with a low outlet temperature for the water. Each case must, however, be considered in relation to the conditions special to it. Thus, for instance, if water is scarce and cooling towers must be adopted, it will often be found that long condensers with small quantities of water are desirable. It must be remembered that a large quantity means a small temperature rise and therefore a correspondingly small temperature reduction in the cooling tower. With surface condensers it is usual to use 50 times as much cooling water as the full-load steam consumption of the turbine.

Example III.—One of the most important factors in condenser design is the inlet temperature of the condensing water.

Vacuum constant at about 27·5in. or 28in.

Tube temperature = 90°.

Outlet temperature = 85°.

Inlet temperature variable.

Curve VII. (Fig. 148) shows how the length of the tubes decreases as the inlet temperature rises. Curve VIII. shows the increase in the tube surface, and Curve IX. the increase in the number of tubes and quantity of

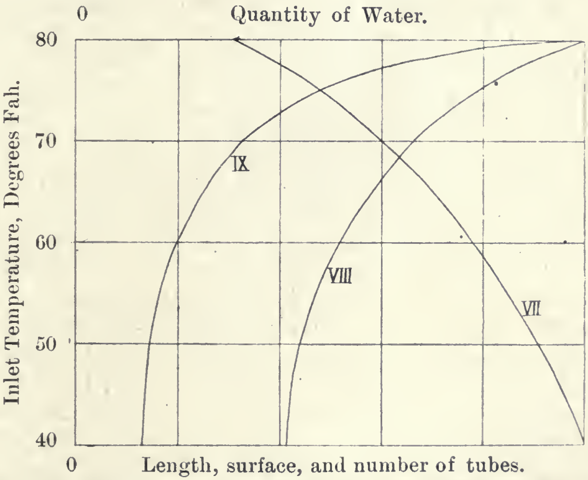


FIG. 148.—EFFECT OF INLET TEMPERATURE ON CONDENSER PROPORTIONS. VII. = length; VIII. = surface; IX. = number and water.

water with an increase in the inlet temperature of the water. The great increase in the tube surface and the quantity of water required is very marked, but would be less so with a lower outlet temperature.

As regards *the effect of the velocity* of the water, it appears from the formula that with a given length of tube the rise of temperature falls off only very slightly with an increase in the velocity, so that the quantity of heat transmitted is nearly proportional to the velocity of the water, and hence the surface (or size) of the condenser will be approximately inversely proportional to the water velocity. For given water and tube temperatures, the length of the tubes will be nearly constant, being slightly

increased with an increase in the velocity. The total work necessary in order to force the water through the condenser will therefore be roughly proportional to the velocity. For instance, suppose that a 1,500 kw. turbine takes 20lbs. of steam per kilowatt hour at full load and that the auxiliaries consume 5 per cent. of this amount, or 1,500lbs. per hour—the consumption of steam (or power) by the auxiliaries is found to vary very little with the load—and that of this amount the circulating pump takes 1,000lbs. Suppose that the velocity of the circulating water is doubled. The size of the condenser will be nearly halved, bringing down the cost of the condensing *equipment* from £1,800 to, say, £1,500. The circulating pump will now take about twice its old quantity of steam or, say, 2,000lbs. If we assume coal to be at 10s. a ton, and that it evaporates 7lbs. of water per pound of coal, we see that for each 1 per cent. of load factor the extra cost of fuel for the circulating pump will be £2·8 per year. The reduction in charges on the condenser equipment (at 12 per cent.) will amount to £36 per year. Hence, the modified condenser with its high-velocity circulating water is the more economical of the two for load factors *up to* about 12 per cent. A little consideration will show that within limits this limiting power factor is roughly independent of the velocity. That is to say, the higher the velocity the greater the economy up to a certain load factor which depends on the costs of fuel and condenser equipments. In the above remarks we have used the term "load factor" as meaning the percentage of *time* during which the plant is running and not in its usually accepted sense. Perhaps "time factor" would express our meaning better.

Announcement has been made by Prof. Weighton* that if the body of a surface condenser be divided into several sections and each section independently drained of its condensed steam, the efficiency of the condenser can be doubled. In other words, the size of the condenser itself can be reduced by 50 per cent. if the condensed steam from the upper portion of the condenser is prevented from falling on to the tubes below it.

Whilst the magnitude of the improvement possible in practice may be much less than this—the writer has

* Institute of Naval Architects, April, 1906.

has not yet (April, 1906) had an opportunity of studying the paper—it is quite reasonable that some improvement might be attained by thorough systematic drainage. A film of water lying practically stagnant on the tubes adds one more surface and a layer of non-conducting material to the path of the heat removed from the steam in condensation. It is conceivable that reducing the thickness of this film might assist the flow of heat somewhat, although a doubling of this rate of flow is very great improvement—if it can be attained in practice.

If this improvement can be realised in practice it will be specially valuable in turbine work.

Power Required by the Auxiliaries.—The power consumed by the auxiliaries varies considerably. The equivalent steam consumption (whether steam or electrically driven) as a percentage of the steam consumption of the main engines is usually from 50 to 100 per cent. greater than the power consumption. This is important, as it is the cost at the coal pile which matters. The higher the vacuum the larger the auxiliaries. These usually consist of

(1) Circulating-water pump,
(2) Hot-well pump,
(3) Dry-air pump to remove the gases from the condenser.

As a rule, each pump is separately driven. For medium and low vacua the hot-well and dry-air pumps are combined into one—the ordinary well-known air pump. Some idea of the variation in the power of the circulating pump with the vacuum may be obtained by noting the relative tube surfaces. The size of the pump will, other things being equal, vary nearly as the tube surface.

In ordinary reciprocating engine practice—say—26in. of vacuum—it is usual to allow about 1sq. ft. of tube surface for each 12lbs. or 14lbs. of steam condensed. This would correspond to about 1·8sq. ft. per kilowatt for a good turbine. Now it is usual in present-day turbine practice to allow from 3·5sq. ft. to 4sq. ft. of tube surface per kilowatt, and even more, the vacuum being usually about 27·5in. or 28in. With a barometric condenser this variation in the size of the circulating pump with the vacuum is not nearly so marked.

In order to reduce the size of the air pump, Mr. Parsons has introduced his vacuum augmentor (Fig. 149). The condensed steam and the gases are taken to the air pump by separate paths. The gases pass through a small auxiliary condenser—containing about one-twentieth as much tube surface as the main condenser. Previous to reaching this auxiliary condenser, a small steam jet is introduced which forces them towards the air pump, and is said to so compress them that with a 26in. vacuum at the air pump, the vacuum in the main condenser will be 27·5in. or even 28in.; the jet taking

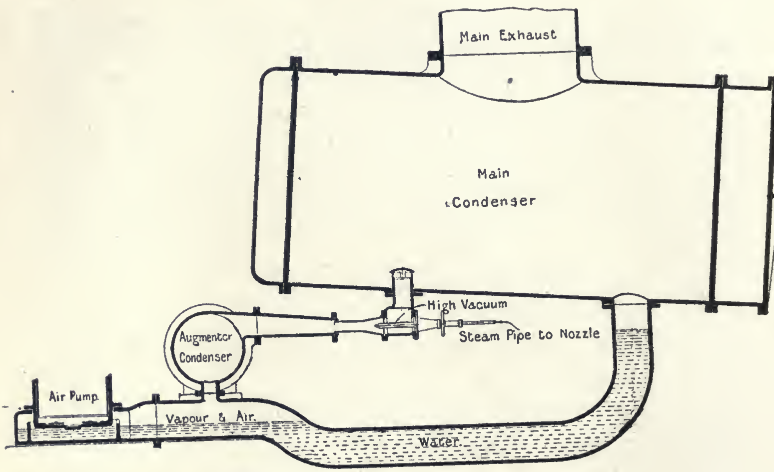


FIG. 149.—ARRANGEMENT OF PARSONS VACUUM AUGMENTOR.

about 1·5 per cent. of the full-load steam consumption of the turbine. This device does not diminish the size of the condenser or the circulating pump, and in view of the fact that it renders the condenser somewhat more expensive and consumes more steam its value (commercially) is not fully proved. Fig. 150 shows the results of tests on a 1,500 kw. turbine with and without the augmentor. The results without the augmentor would be better with a larger air pump, of course.

As regards the power required by the auxiliaries, Mr. Bibbins states that the power taken by the steam-driven auxiliaries of a 500 kw. Westinghouse turbine at Johnstown, Pennsylvania, amounted to rather less than

5 per cent. at 100 kw. load, and about 2·5 per cent. at 500 kw. The percentage steam consumption would be from 1·5 to twice this amount or even much more.

The vacuum was about 27·5in. (reduced to a 30in. barometer). The exhaust steam from the auxiliaries was used in an open feed-water heater, thus recovering some of the waste heat. Mr. Bibbins also gives the power

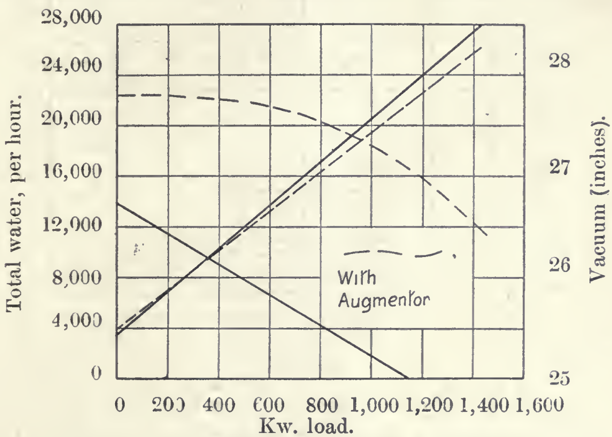


FIG. 150.—PARSONS TURBINE, 1,500 KW., AT SHEFFIELD.

consumption of the auxiliaries of a 2,000 kw. Curtis turbine at the St. Louis Exposition. The auxiliaries were driven by induction motors, and the conditions were unfavourable because of the high temperature of the condensing water. The results are given in Fig. 151. It will be seen that the power of the auxiliaries as a percentage of the output of the turbine varies from 14 per cent. at half-load down to 7 per cent. at full load. It is evident that at about quarter-load the power required by the auxiliaries would be quite large. In a test by Prof. Ewing on a 500 kw. Parsons turbine driving its own air and circulating pumps, it appeared from comparisons with a test where the auxiliaries were separately driven that the auxiliaries required pretty much the same power at all loads, and that this power was about equal to 3 per cent. of the full-load output. The vacuum was about 27·5in.; barometer 30in. The power required by the auxiliaries, especially the circulating pump, will undoubtedly increase fairly rapidly with an increase in the vacuum required.

Tests on a 500 kw. Curtis turbine in America carrying a rapidly-varying load at a power station showed that the auxiliaries were consuming about 3·5 per cent. of the total output at full load and 7·3 per cent. at half-load.

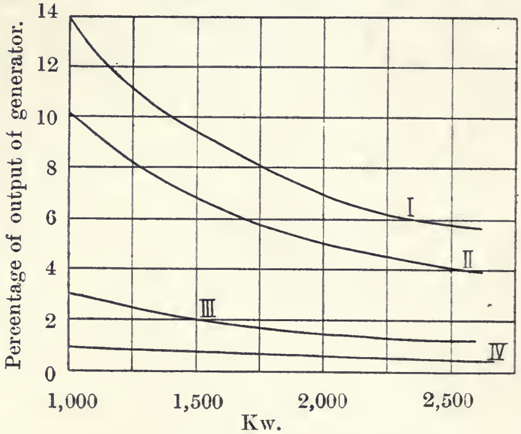


FIG. 151.

I. = total. III. = air pump.
II. = circulating pump. IV. = hot-well pump.

Mr. Sparks has recently tested a 1,500 kw. Curtis turbine at Wandsworth. The auxiliaries were all driven by 2-phase motors. The particulars of these auxiliaries are given below.

Auxiliary.	Kw. Used by Auxiliary at		Per Cent. of Actual Turbine Output at	
	Quarter Load.	Full Load.	Quarter Load.	Full Load.
Edwards' air pump, 15 kw. motor, 2-foot-step pumps, 2 kw. motors	5·75	6·75	1·54	0·45
Centrifugal pump, 30 kw. motor	21·5	36	5·73	2·40
Total	27·25	42·75	7·27	2·85

The condenser has a surface of 4,000 sq. ft., and the vacuum was 29in.

The auxiliaries for the 1,500 kw. Curtis turbines of the Yorkshire Electric Power Company are electrically

driven; the Edwards 3-throw air pump by a 15 h.p. and the circulating pump by a 43 h.p. motor, or about 3 per cent. of the full-load output of the generator.

Diverging Exhaust Pipe.—The vacuum in the condenser is usually lower than that at the turbine exhaust. It should be possible, however, to reverse this, and have a lower vacuum in the condenser than at the turbine, by suitably increasing the cross-section of the exhaust pipe as we approach the condenser. In this way the kinetic energy in the steam will be reduced and partially converted into pressure energy. A considerable waste into heat will take place, but this of itself is of very little consequence. This method of construction would be particularly applicable to a turbine of the De Laval or single stage Riedler-Stumpf turbine, where the velocity of the steam as it leaves the turbine wheel varies from about 1,000ft. to 1,400ft. per second. In these cases it might be best to receive the steam in specially-constructed compressing nozzles. Suppose the exit velocity to be reduced from 1,200ft. per second to 400ft. per second, the pressure at the turbine exhaust being 1lb. absolute and the dryness $0 \cdot 8$. The loss of kinetic energy is equal to $25 \cdot 6$ B.Th.U. If $9 \cdot 6$ are merely converted into heat, and the rest into pressure energy, the condenser pressure will be $1 \cdot 38$lbs. per square inch absolute, the steam temperature being raised 11° Fah., thus largely reducing the size of the condenser and the air pump without in any way altering the steam consumption of the turbine. If the exhaust velocity is above 1,200ft. per second, a *converging* exhaust pipe may be required to give the above results.

The Best Vacuum.—Turbine engineers usually urge the use of a very high vacuum, but it is doubtful if it pays in the majority of cases, and it is certain that in some existing installations non-condensing turbines would give better commercial results than the present condensing turbines. Interest, depreciation, and the other fixed charges on a condensing plant and all the other things accessory to it are always running on; whereas the reduction in the fuel cost which they were installed to accomplish is only secured when the turbines are delivering power.

Fig. 152 is from a paper by Mr. J. R. Bibbins,* and illustrates the relative costs of complete condensing plant (consisting of surface condensers, dry-air pumps, hot-well pumps, piping, valves, &c.), designed to give

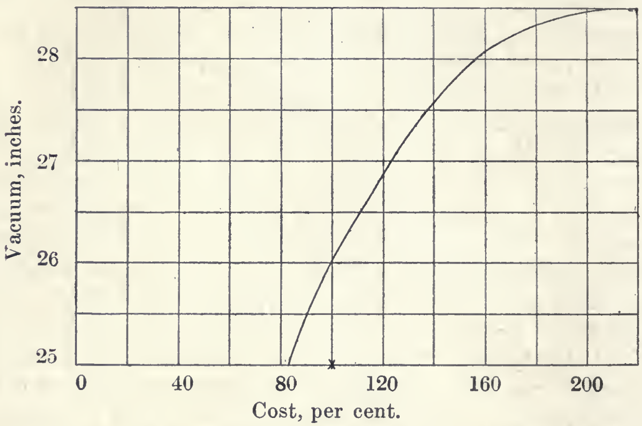


FIG. 152.—RELATIVE COSTS OF CONDENSERS FOR DIFFERENT VACUA.

different vacua. It will be noticed how much more costly it is to improve the vacuum 1in. at 27·5in. vacuum than at 26in.; three times as expensive (in plant), in fact. Nor does this cost include all that ought to be put down as part of the cost of the condensing equipment. All costs which are due to the extra vacuum ought to be charged against it. They include the following; cost of—

(1) Condensing plant proper (pumps, condenser, piping, &c.).
(2) Basement or condensing room.
(3) Extra land required by condensing equipment, cooling tower, &c.
(4) Cooling tower (if any).
(5) Extra rates, taxes, insurance.
(6) More expensive turbine or engine.

As against this there will, where large variations in the vacuum are being considered, be a reduction in the cost of the boiler installation, coal bunkers, &c.

* "Steam Turbine Power Plants," by J. R. Bibbins. St. Louis Convention of the American Street Railway Association, October 12th, 1904.

In order to determine the best vacuum we must obtain a curve (Curve I., Fig. 153), showing the *rate of increase* of the fixed charges on this condensing equipment with the vacuum. Then we determine curves showing the *rate* of decrease of the fuel cost (from which has been deducted the extra cost of water, oil, and stores generally) for all load factors and various prices of fuel. The rate

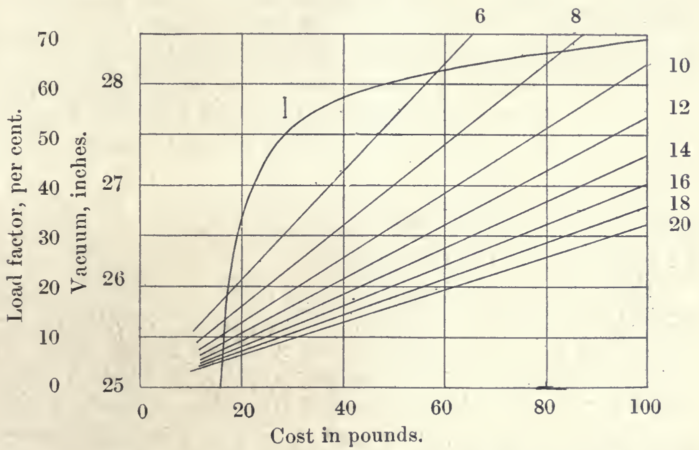


FIG. 153.—BEST VACUUM FOR TURBINE ; SURFACE CONDENSER, NO COOLING TOWER.

of increase in the cost of the condensing equipment is obtained by drawing a curve showing the relation of the total fixed charges (interest, depreciation, attendance, &c.), on this equipment to the vacuum. The inclination of the curve at any vacuum gives us the rate of increase at that vacuum. This is not the same thing as the actual increase between that vacuum and the next inch, but is always less, owing to the fact that the rate of increase is continually increasing with the vacuum.

For instance, the rate of increase in the charges per inch of vacuum at 27·5in. is £28·7, whereas the extra cost of an increase from 27·5in. to 28·5in. is £67.

As regards the rate of decrease in the fuel cost with increase in vacuum, the results of tests vary somewhat. These tests will be referred to in the next chapter. For the present we shall assume a uniform decrease in the steam consumption of 0·56lb. per kilowatt hour per inch of vacuum.

This saving is the *net* saving, and is intended to include an allowance for the extra water, oil, &c., required at high vacua. The saving is greater at light loads, so far as steam consumption goes, but as the boiler losses then become somewhat higher we can say that the saving at the coal pile per kilowatt hour is independent of the load.

As regards reciprocating engines, data are scarce, but in view of the results of theory and the general trend of experience, it seems probable that it varies from about 0·37lbs. per kilowatt hour (per inch of vacuum) at 16in. of vacuum, to about 0·19lb. per kilowatt hour at 28in. of vacuum. In what follows, we shall assume these figures to be correct.

The above figures are intended to serve for a large unit of about 1,000 kw. Somewhat greater saving will be obtained with small units.

Prof. Weighton experimented with a triple-expansion engine at different vacua. There was practically no reduction of steam consumption after about 26in. of vacuum. Mr. Mark Robinson, in the discussion on the paper by Prof. Weighton, stated that the saving in steam consumption was one per cent. per inch of vacuum with the Willans central-valve engine.

In a paper by R. W. Allen on "Surface-condensing Plants,"* the results of tests on reciprocating high-speed engines showed a gain of from 0·6 to 1·0 per cent. at high vacua, and rather more at quite low vacua.

If the condensed steam is returned to the boilers, high vacua necessitate a considerable increase in the coal consumption due to the low temperature of the condensed steam. This increased coal consumption seems to vary from about 0·25 per cent. at 15in. of vacuum to about 1 per cent. at 27in. of vacuum. In view of this fact, it seems improbable that, where the condensed steam is returned to the boilers, there can be any reduction in the coal consumption for vacua greater than about 26in. with reciprocating engines. This does not hold where an injection condenser is used, or where fresh water is used to feed the boilers. In the calculations which follow, no allowance for the temperature

* Proceedings of Institution of Civil Engineers. Vol. CLXI. 1904-5. Part III.

of the feed water—whether for turbines or reciprocators—has been made. If, however, a surface condenser returning its condensed steam to the boilers is used, then the best vacuum as determined from the diagrams—or similar ones drawn to suit the particular conditions—should be reduced by about ½in. or more in the case of a reciprocator.

If coal costs 10s. a ton of 2,240lbs., and each pound of coal evaporates 7lbs. of water, then the fuel cost per year for a 1,000 kw. turbine, for each 1 per cent. of load-factor,* is decreased by

$$\frac{365 \times 24 \times 560 \times 0{\cdot}5}{2{,}240 \times 7 \times 100}$$
$$= £1{\cdot}56.$$

Thus an increase in the vacuum from 26in. to 28in., at a load factor of 20 per cent., would reduce the fuel cost per year by £62·5. If coal is not at 10s. a ton we have to alter the saving in proportion to the price of coal.

For a 1,000 kw. reciprocating engine, the reduction in fuel cost per inch of vacuum for each 1 per cent. of load factor has the following values in £ per year (coal at 10s.), at the vacua named.

In. vacuum..	28	27	26	25	24	23	22	21
Saving ...	·53	·57	·61	·65	·69	·73	·78	·82

From the above data we can determine the rate of decrease of coal cost at all load factors and prices of fuel.

For a turbine this rate of decrease is, roughly, independent of the vacuum, but is proportional to the price of coal and the load factor.

We will illustrate this by a few examples.

Case I.—1,000 kw. turbine: *Net* saving in steam per kilowatt-hour per inch of vacuum = 0·56lb.

Cost of surface-condensing plant for 28in. vacuum = £1,200; costs at other vacua obtained from the curve in Fig. 152 as given by Bibbins. The fixed charges,

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* Load factor = $\frac{\text{Yearly output (in units).}}{\text{Capacity for yearly output.}}$

This is not quite the usual definition of load factor, which is the ratio of average to maximum load. For our purposes the ordinary definition is useless, seeing that *all plant installed*, whether used or not, has to pay fixed charges. This point is of importance, as the load factor calculated by our method is seldom more than about half or two-thirds the ordinary load factor.

interest, depreciation, rates, &c., are taken at 12 per cent. of the capital outlay.

In Fig. 153, Curve I. shows the rate of increase in these fixed charges at the different vacua. The curves marked 10, 12, &c., show the rate of decrease in the fuel cost at all load factors for coal at 10s., 12s., &c.

Use of the Diagram.—When the rate of decrease in the fuel cost just balances the rate of increase in the fixed charges, we have arrived at the best vacuum. If we go to higher vacua the *further* investment in condensing plant does not pay for itself. It is important to note that we do not go on until the total reduction in fuel cost is equal to the total increase in the fixed charges. As this point is very important, we may take an illustration from business life. A man has an engineering works in which he has invested £10,000. His net profit, after paying all charges, is £1,000. He invests another £1,000 and gets an additional profit of £50. He invests another £1,000, but finds it difficult to keep his enlarged works fully employed, and his additional profit is only £10. Suppose he invests still another £1,000 and finds that he makes on this a net loss of £10. Clearly he has invested too much in the business, although his net total profit is still £1,050. These figures are presented in tabular form below :—

Last Investment.	Net Profit on Same.	Total Capital.	Total Net Profit.
10,000	1,000	10,000	1,000
1,000	50	11,000	1,050
1,000	10	12,000	1,060
1,000	loss 10	13,000	1,050

We must stop investing capital in condensing plant when the net gain produced by a *small* increase of vacuum is nothing. This is what the diagram enables us to do. For instance, suppose load factor is 20 per cent., and coal costs 8s. a ton. On the straight line marked 8 in Fig. 153, at the load factor 20, we see that the rate of decrease in the fuel cost is £250. The increase in the fixed charges is £25 for a vacuum of 27·3in. Then this is the best vacuum. We see from the diagram

that with coal at 10s. it does not pay to run condensing at all if the load factor is less than about 8 or 10 per cent. With coal at 20s., this limiting load factor is about 4 or 5 per cent.*

Case II.—In the last example the cost of the condensing plant was such as to allow nothing for the accessories (condenser room, &c.), required by the plant, and may be taken as representing about the cheapest surface-condensing equipment practicable. In this example we shall assume certain other costs in addition to those of the condensing plant proper, as follows :—

1,000 kw. turbine. Steam consumption as in Case I. Cost of condensing plant proper (condenser, pumps, piping, &c.), as in Case I. (£1,200 at 28in. of vacuum).

Cost of cooling tower, basement tank, foundation, and fan varying from £900 (12s. per horse-power), at 15in.† vacuum, to £1,200 (16s. per horse-power), at 28in.

Cost of extra buildings, lands, &c., varying from £325 at 25in. vacuum up to £400 at 28in. vacuum.

Fixed charges taken at 12 per cent. of the capital cost.

Fig. 154 illustrates this case. The method of using the diagram is precisely as for the previous example. We

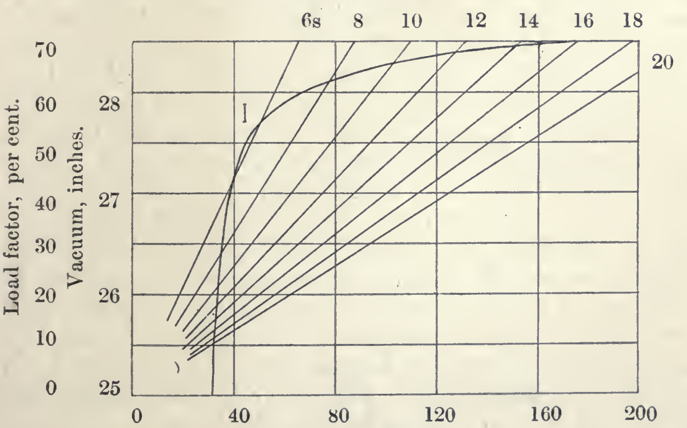


FIG. 154.—BEST VACUUM FOR TURBINE; SURFACE CONDENSER AND COOLING TOWER, &C.

* See note at end of this chapter, regarding the influence of the boilers.

† This is perhaps rather too favourable towards the cooling tower at very high vacua.

see from the diagram that with a coal price of 10s., the limiting load factor below which it does not pay to run condensing is about 14 or 15 per cent.

Case III.—Although this book is primarily concerned with turbines yet it may be advisable to just indicate the method of drawing the diagrams for determining the best vacuum for reciprocating engines. With reciprocating engines the reduction in the steam consumption is less (per inch of vacuum) at high vacua than at low. Full data are not available, but the results of a careful study of indicator diagrams and the general trend of practical experience suggest that if we take a uniform variation in the net saving, from 0·37lbs. of steam at 16in. of vacuum (per kilowatt hour) down to 0·19lb. at 28in. of vacuum we shall not be very far wide of the mark. The cost of condensing plant for a reciprocating engine is considerably more than for a turbine, on account of the greater quantity of air finding its way into the exhaust, air being a great enemy to successful condensation, as well as requiring a larger air pump. The cost of the condensing plant proper has been taken at 15 per cent. higher than for a turbine. The other costs remain unaltered. In this example we will consider a 1,000 kw. set with only a condensing plant proper, no cost being allowed for extra land, cooling tower, &c. Now, owing to the fact that the rate of decrease in the fuel cost varies with the vacuum as well as with the load factor, we must modify our diagram somewhat. We know that the saving in fuel is proportional to the cost of the coal, the load factor, and the vacuum. The straight lines marked 400, 800, &c., in Fig. 155 show the rate of decrease in the fuel cost at the different vacua when the products of the price of coal in shillings per ton and the load factor per cent. are respectively 400, 800, &c. Thus, line 400 might represent coal at 10s. and a 40 per cent. load factor, or coal at 16s. and a 25 per cent. load factor. For each one per cent. of load factor and shilling per ton for coal the saving in fuel cost at 28in. of vacuum will be (per year)

$$\frac{365 \times 24 \times 0{\cdot}19 \times 1{,}000 \times 0{\cdot}01 \times 1}{2{,}240 \times 7}$$

$$= £0{\cdot}106.$$

At 25in. of vacuum it will be £0·181. These are the rates of decrease in fuel cost under the conditions mentioned.

Fig. 155 illustrates this case. It will be seen that with coal at 10s., the least load factor at which it will pay to run condensing is about 15 per cent., and then a very low

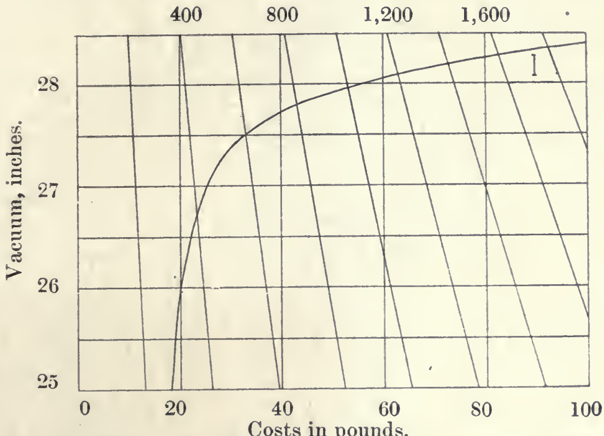


FIG. 155.—BEST VACUUM FOR RECIPROCATING ENGINE; SURFACE CONDENSER, NO COOLING TOWER.

vacuum (about 20in.) is all that is desirable. If we want to know the best vacuum for a load factor of 40 per cent., and coal at 15s., then we look along the line marked 600, and where it cuts the curve showing the rate of increase in the fixed charges (Curve I.) marks the best vacuum, 27·5in. in this case.

Case IV.—Fuel consumption as in Case III. Cost of condensing equipment as in Case II. Fig. 156 illustrates this case, the explanation of the diagram being as for Fig. 155 of the previous case.

Although we have assumed a unit of 1,000 kw. capacity, yet within wide limits—indeed, for all except the very small units—the above diagrams will stand equally well for all sizes of units. The principles underlying the construction of these diagrams can, of course, be applied equally well to a very large variety of cases, besides those for determining the best vacuum, the best proportions for surface condensers, and other such-like problems.

Condensing Plant for Central Stations.—The load factor in electrical generating stations in England using turbines averaged about 15 per cent., and was in no case more than 16 per cent. At first sight, then, it would appear that these plants ought all to be running either non-condensing or on a low vacuum, depending on the cost of condensing water, coal, and land. However, there is another and

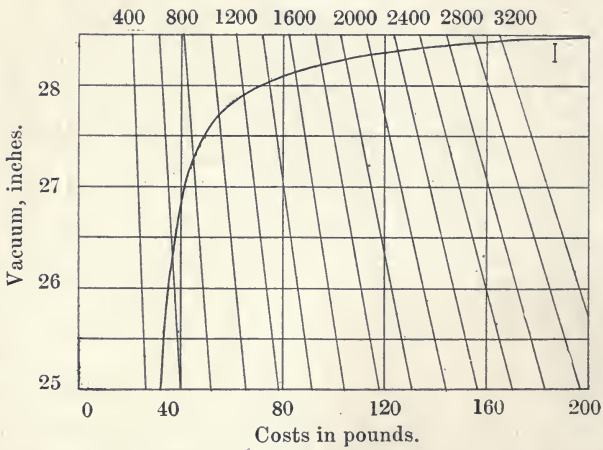


FIG. 156.—BEST VACUUM FOR RECIPROCATING ENGINE; SURFACE CONDENSER AND COOLING TOWER, &C.

better solution of the question. For instance, suppose that in the station there are four similar turbines with a combined load factor of 15 per cent. During most of the day one turbine alone will be running. When the peaks come on one or two of the others will be put in, and during the winter peaks all four will be put on. The individual load factors might be, say,—

	Per cent.
Turbine (1)..	38
,, (2)..	10
,, (3)..	7
,, (4)..	5

Then, under usual conditions, turbines (2), (3), and (4) will have to be non-condensing, whilst turbine (1) would certainly be equipped with a condenser, probably operated at a high vacuum. It may be objected that the same engine does not always take the day load. If, however, two—or all four, if necessary—of the turbines

are connected to the condenser, then whichever turbine is taking the day load would be running condensing. The other three (if running) would be exhausting into the atmosphere; although it would probably be more economical to couple them up to the condenser. This, of course, would greatly reduce the vacuum, but it would be only for a short time. Under these conditions, the single condenser would be condensing the steam for two, three, or even four turbines. In any case, however, a condensing equipment for only one or at most two turbines ought to be provided. Indeed, in many cases it would undoubtedly be best to provide only one condensing equipment, but to turn the exhaust steam from the others into the condenser.

For instance, suppose we have the following results (one condenser):—

Number of Turbines Condensing.	Vacuum, About.	Lbs. per Kilowatt Hour.
1	28	20
2	26·7	21
3	25·4	22
4	24	23
0	0	30

Case I.—No. 1 only condensing. Steam per year is proportional to—

Turbine (1)	38×20	=	760lbs.
„ (2)	10×30	=	300lbs.
„ (3)	7×30	=	210lbs.
„ (4)	5×30	=	150lbs.
Total			1,420lbs.

Case II.—All use condenser when running. Steam consumption per year is proportional to—

Turbines (1)	$28 \times 20 =$	560lbs.
„ (1) and (2)	$2 \times 3 \times 21 =$	126lbs.
„ (1), (2), and (3)	$3 \times 2 \times 22 =$	132lbs.
„ (1), (2), (3), & (4)	$4 \times 5 \times 23 =$	460lbs.
Total		1,278lbs.

Q

Thus clearly since the fixed charges will be nearly the same in both cases (a little greater in Case II, because of the extra piping and valves), the second method is the more economical of the two by about 10 per cent. so far as steam consumption goes, and is still more economical commercially, when we remember the somewhat reduced boiler capacity required by the station.

In our diagrams no allowance was made for the increased cost of the boilers at low vacua. For a variation of 4in. or 5in. in the vacuum, the effect on the boiler plant will be quite small; but as between condensing and non-condensing there will be a marked difference in the initial cost. For very small load factors and cheap fuel it would therefore be advisable to draw special diagrams extending to no-vacuum conditions, in which allowance is made for the extra cost of the boiler plant at low vacua.

Of course conditions vary considerably, and it may often be advisable to draw special diagrams suitable to the particular conditions rather than rely on the necessarily somewhat general diagrams given in this chapter.

In general, a combination of turbines taking the main load and operating continuously at high vacua, with reciprocating engines for assisting at the peaks and running with low-vacua or non-condensing, will give the most economical results.

Chapter XI.

STEAM TURBINE PERFORMANCE.

In this chapter it is proposed to point out some of the factors affecting the steam consumption of turbines, and to give particulars of some tests.

Effect of Vacuum.—A steam turbine makes good use of a high vacuum; better indeed than a reciprocating engine. Hence, seeing that over the whole range of

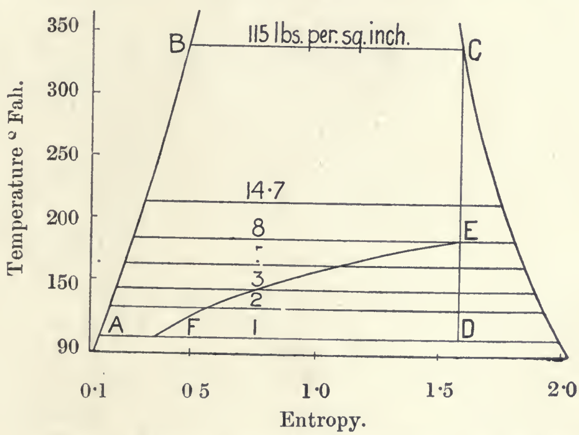


Fig. 157.—Effect of Vacuum on Turbines and Reciprocators.

pressure from boiler to condenser, the turbine and reciprocator come out very nearly even, it follows that the reciprocator must be the more efficient at the higher pressures. The reason is at once made clear if we refer to the entropy diagram in Fig. 157. There are sketched the work diagrams for a reciprocator and a turbine between pressure limits of 115lbs. and 1lb. per square inch, absolute. On account of the great volume of the low-pressure steam the reciprocator cannot handle it, and has to release at, say, 8lbs. absolute. The theoretical works for the

reciprocator and the turbine are then seen to be represented by the areas A B C E F and A B C D, which have the values in British thermal units of 248 and 300, the latter being 21 per cent. better than the former. The lower the initial pressure, the more marked does this difference become. The above are the theoretical values, but they show clearly that a low-pressure turbine must be more efficient than a low-pressure reciprocator. Hence it may be desirable in some cases to replace the low-pressure cylinder of the reciprocating engine by a turbine receiving the exhaust steam from the former at about 10lbs. absolute. This combination would have special advantages on board ship. It would admit of perfect

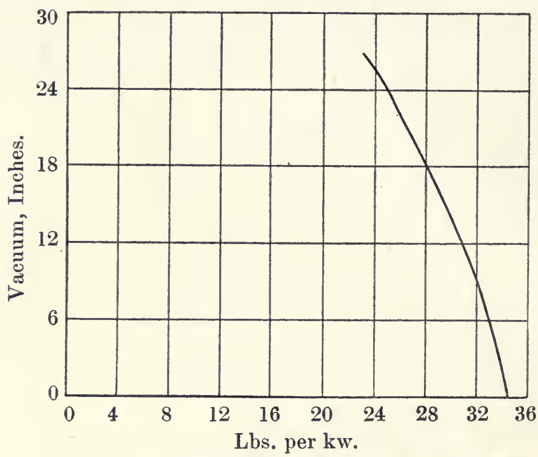


FIG. 158.—EFFECT OF VACUUM, PARSONS TURBINE.

ease in reversing the propeller, and would be from 10 to 15 per cent. (or even more) more economical than the all-turbine or all-reciprocator drive.

A glance at the entropy diagram will show that the economy of a turbine increases rapidly with high vacua, and these results are fully borne out in practice. Fig. 158 gives the results of tests on a 300 kw. Parsons turbine at full load with very different vacua. It will be noticed that an increase in the vacuum is conducive of greater economy at high than at low vacua. Tests on the Westinghouse-Parsons turbine seem to point to a uniform decrease in the steam consumption for each extra inch of

vacuum. Similar results are obtained with the Curtis and other turbines.

The following table (Table XIV.) illustrates the percentage gain in economy per inch of vacuum at various vacua. The theoretical results were not determined with any very great degree of accuracy, because so much depends on the initial conditions—pressure and superheat—which in this case were 160lbs. (absolute) pressure and dry saturated steam. The close agreement between theory and practice is very suggestive.

TABLE XIV.

Inches Vacuum		29	28	27	26	25	16	Superheat.
Gain per cent.	Parsons ...	6	5	4	3·5	3	2	0
	Westinghouse	—	3·14	3·05	2·95	2·87	—	0
	Curtis	5·4	5·1	4·8	4·6	4·2	—	—
	Theoretical ...	9	5·2	4·4	3·7	3	2·4	0

Unlike the effect on a reciprocating engine, a high vacuum used with a turbine does not lead to any initial condensation.

The reduction in steam consumption per kw. hour for each extra inch of vacuum is usually about half-a-pound. According to a curve given by Emmet, double this gain is attained in Curtis turbines, but so far as the present writer is aware there are no published figures bearing out this claim.

Effect of Superheated Steam.—Whether or not superheated steam is conducive of economy from a commercial point of view, we will not attempt to discuss. The use of superheated steam is now very common, and is found to lead to economy of water consumption, and of heat consumption at the turbine. The saving in steam consumption is usually reckoned at about 1 per cent. for each 10° Fah. of superheat. Emmet gives the decrease in steam consumption as being about 1·7lbs. per 100° of superheat. This is illustrated by Fig. 159, which gives the results of tests on a 500 kw. Curtis turbine, the load being nearly constant at about 525 kw.

In Fig. 160 we have the results of tests by Messrs. Dean and Main on a 400 kw. Westinghouse turbine, showing the gain due to the use of superheat at various

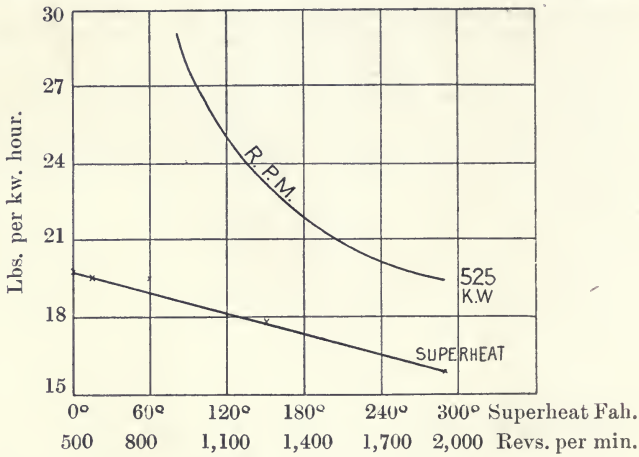


FIG. 159.—TESTS ON 500 KW. CURTIS TURBINE.

loads. It will be seen that the decrease in the steam consumption is about 2·1lbs. per kilowatt hour per 100° of superheat at all loads. According to some tests on a

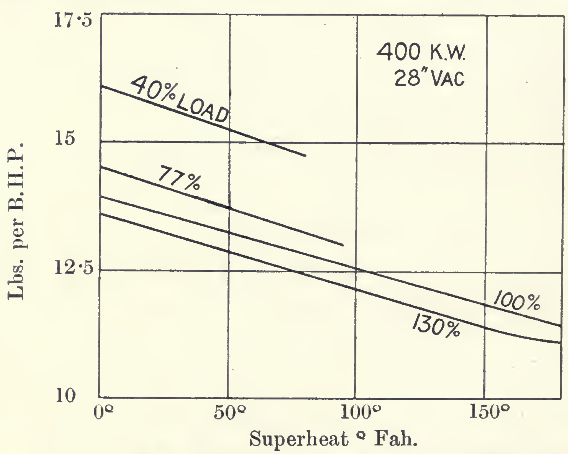


FIG. 160.—WESTINGHOUSE TURBINE. TESTS BY DEAN AND MAIN.

1,500 kw. Westinghouse turbine a reduction in the steam consumption of 1·55lbs. per kilowatt per 100° of superheat was obtained.

The gain in steam economy seems to be due partly to thermal and partly to mechanical reasons. The gain on account of thermal reasons is easily understood. In the first place each pound of steam contains more heat when superheated than when saturated, and hence the turbine requires less steam ; and also since the heat of superheat is received at a higher temperature than that of the saturated steam, the turbine ought to make a more efficient use of it, thus again reducing the steam consumption. For instance, with steam at 155lbs. per square inch (absolute), taking the specific heat of superheated steam as uniformly equal to $0 \cdot 5$, the relative theoretical steam consumptions at 0°, 50°, 100°, and 150° of superheat ought to be 100, $97 \cdot 5$, 95, and 92 per cent., or a decrease in steam consumption of only 5 per cent. for 100° of superheat, the vacuum being 1lb. absolute. It is quite possible—indeed probable—that the specific heat of superheated steam is greater than $0 \cdot 5$, but even so it is clear that the thermal reasons alone will not account for the decrease in steam consumption of about 10 per cent. for 100° of superheat, as actually observed ; there must be some other reason.

This other reason seems to be that the friction of superheated steam is less than that of saturated steam at the same pressure, as we saw in a previous chapter. We saw there that the reduction in friction with an increase of superheat was less the greater the superheat, so that the gain from this cause will be less proportionately at high than at low values of superheat. On the other hand, as the previous numerical example has shown, the proportional thermodynamic gain is greater the higher the superheat ; so that the two influences combined will tend to give a uniform decrease in steam consumption with increase in superheat, as is actually found to be the case in practice.

The gain due to the use of superheated steam in a reciprocating engine seems to be partly due to reduced interchanges of heat between the steam and the metal, due partly, it is thought, to the retarding effect of the dry walls, although the temperature range is increased ; partly due to thermodynamic reasons, just as in a turbine, but chiefly due to the reduction of leakage. This leakage

takes place mainly as water, but is not, as so many people seem to think, blown through the very small space between the valve and its seating by the difference in steam pressure on the two sides of the valve, although this may assist the action. What occurs seems to be something of this nature. The hot steam at admission meets the cold metal surfaces (cooled by the low temperature of the exhaust steam) and some of it is condensed, thus forming a layer of moisture on the valve and its seat as well as on the piston rod and the rest of the exposed surface. The valve now moves so as to carry this wet surface into contact with the exhaust chamber, where, owing to the lower temperature, it is evaporated. As the valve moves back, part of the valve seat

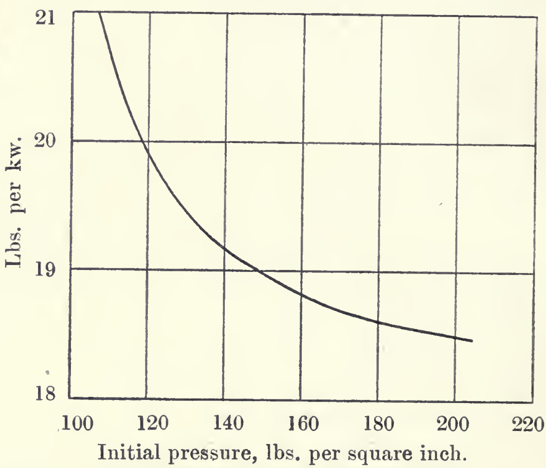


FIG. 161.—600 KW. CURTIS TURBINE, 1,500 REVOLUTIONS PER MINUTE. 28·5″ VACUUM. EFFECT OF INITIAL PRESSURE.

previously in contact with the hot cylinder steam is brought into contact with the exhaust and its moisture evaporated. The water on the metal surfaces is carried over in the same way that oil on the crosshead bearing surfaces is carried along to different parts of the slide bars. It is the viscosity of the water and not the lack of it which assists the leakage.

Now, nothing of this kind can take place in a turbine. The presence of moisture on the metallic surfaces will, however, probably increase the friction, apart from the

density of the steam. It is for this reason that a steam jacket has been suggested for use on a turbine. It is very doubtful whether a steam jacket could keep the blades (and particularly the moving blades and drum) dry, so that, except possibly in the case of a turbine using a blast governor on light loads, it is very doubtful indeed whether a steam jacket would be of any use except—and this point is worth considering—to maintain the cylinder at a uniform temperature and thus reduce the troubles from unequal expansion; and even here the advantage is very problematical.

Initial Pressure.—Steam turbines seldom operate at pressures much higher than 190lbs. per square inch

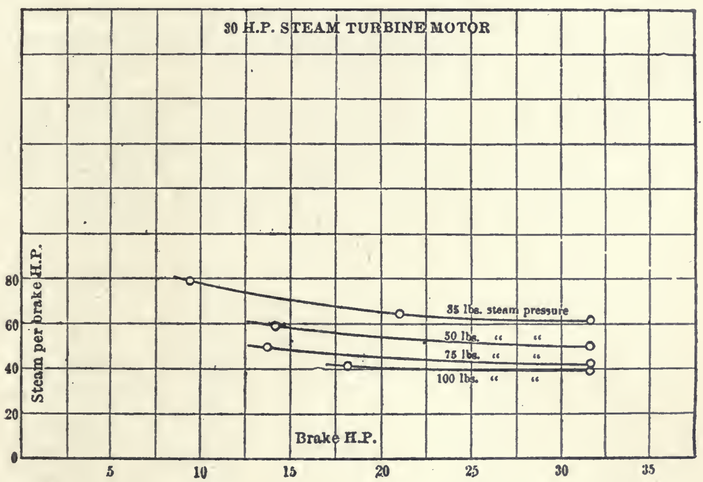


FIG. 162.—EFFECT OF INITIAL PRESSURE ON ECONOMY OF DE LAVAL TURBINE.

(absolute), and as a rule about 160lbs. per square inch is used. Fig. 161 is taken from a paper by Mr. Emmet, and gives the steam consumption of a 600 kw. Curtis turbine at 1,500 revs. per minute and 28·5in. vacuum at various initial pressures (absolute). In view of the greater cost of a high-pressure plant, it is clear that too high a pressure is not economical.

Fig. 162 shows the results of tests on a 30 h.p. De Laval steam turbine, when supplied with steam at the pressures marked on the curves. The turbine was

running non-condensing, the number of nozzles admitting steam to the wheel being varied to suit the load. If the turbine had been running condensing, the effect of the initial pressure on the steam consumption would be much less marked. As it is, it will be noticed how much greater effect each extra pound of initial pressure has at low than at high initial pressures.

Steam Consumption Figures: Effect of the Size of the Turbine.—The larger the turbine the more economical will it be. The efficiency of very small turbines of the reaction type is apt to be low on account of the large friction losses, the low peripheral velocities, and the large leakage over the ends of the blades. This last cause is not so important for an impulse turbine. It is probable that except where there are special reasons to the contrary, turbines of less than 200 kw. or even 400 kw. are not as economical as a corresponding reciprocating set.

The following table (Table XV.) is compiled from the results of tests on Parsons' turbines at full load, and shows how the economy depends upon the size. The results have been reduced to no-superheat conditions.

TABLE XV.

Size (kw.)	Load (kw.)	Vacuum.	Pressure.	Lbs. per kw.
50	52·7	28·0	126	28·0
100	123	27·8	129	26·9
100	122	27·8	134	25·3
135	138·3	27·15	140	25·05
200	204·4	27·71	134	22·4
300	312·1	27·8	150	21·1
350	359·5	27·82	150	22·1
500	501·9	27·4	145	21·3
1,500	1,442	27·45	196	19·4
1,500	1,585	27·52	128	19·8
3,000	2,995	27·0	138	18·2

Effect of the Turbine Speed.—Consider one stage of a turbine and let

v = velocity of steam parallel to wheel rim.

V = velocity of wheel.

Then the work done—neglecting all losses—is

$$W = k\ (Vv - V^2)$$

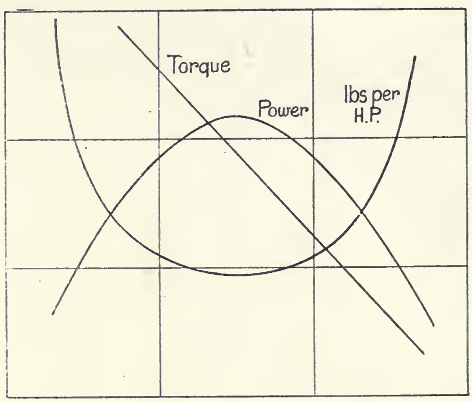
where k is a constant.

If we plot a curve showing the relation of W to V, we obtain what is called a power parabola, v being assumed constant. Where there are many stages, the above equation must be modified somewhat if it is to give the relation of work to speed. When the speed (V) is too small, the exit velocity from the first wheel will be too high, and of this extra kinetic energy some will find its way into the steam entering the second set of moving blades, thus giving for the second and all succeeding sets of moving blades a higher steam velocity v, which will increase the work done somewhat. If on the other hand V is too large, the reverse action will occur, and our formula will indicate too much work. The effect of the speed upon the losses can be estimated very roughly. The eddy losses will be at a minimum when the speed V is that for which the blade angles are most suitable, and will increase as V gets less or greater than this value. The surface friction losses (we are here dealing with a turbine of fixed size and proportions) will be approximately proportional to the cube of the relative (to the surfaces of the blades) steam velocities, and these will slightly increase as V decreases, owing to the greater exhaust velocities. Hence the surface friction will decrease as V increases, though not very rapidly. The friction losses at the high-pressure end partially re-appear as kinetic energy towards the low-pressure end of the turbine. The final results will therefore be somewhat as in Fig. 163, in which one curve represents the power generated (for a constant steam consumption) and the other the steam consumption per unit of power. Fig. 159 is from a test on a 500 kw. Curtis turbine at full load but varying speeds. So far as the curve goes, it is evidently similar to that which we have just deduced. In Fig. 163 we also have a curve of torque T. It will be seen that the torque is a maximum at starting, so that the turbine will start up rapidly and will be very suitable for replying to a sudden demand for speed, such as is sometimes required in ships.

Economy at Different Loads.—We pointed out in a previous article how the steam consumption varied with the load.

The figures given there, however, were based on the most economical load. In the accompanying table (Table XVI.) we give the steam consumption as a percentage of the rated full-load consumption, comparative figures being also given for some representative reciprocating engines, for a gas engine set of the 3-cylinder vertical type using a throttle governor, and also for a small Crossley gas engine.

The two Rateau, the Zoelly, and the 20 kw. De Laval turbines used the throttle governor. The Curtis



Revolutions per minute.

FIG. 163.—EFFECT OF TURBINE SPEED.

and the 200 kw. De Laval turbines governed by cutting-out nozzles. The other turbines were all governed by the well-known Parsons blast governor.* With the exception of the throttle-governed De Laval there is not a great deal to choose between the relative steam consumptions at varying loads, although the nozzle cut-out method seems to give the best results where it is applicable.

The steam reciprocators are at least as economical at light loads—relatively to the full-load consumption—as the turbines, and the larger engines with automatic cut-off are distinctly superior.

The gas engines give comparatively poor results at light loads, particularly so the Westinghouse engine with throttle governor, which gives results very much

* The writer is not certain that the Brush turbines did not use a throttle governor.

inferior to the hit-and-miss of the Crossley engine. Double-acting gas engines might perhaps give somewhat better results.

TABLE XVI.

Per Cent. of Rated Load.	150	125	100	75	50	
Rateau, 1,000 kw.	..	..	100	78	55·6	33·7
Rateau, 500 kw.	..	..	100	77·2	55	32
Westinghouse (U.S.A.), 1,250 kw.	152	122	100	79·5	58·3	38·4
Westinghouse (U.S.A.), 400 kw.	..	..	100	78·5	57	..
Parsons, 3,500 kw.	154	124	100	77·5	55·3	33
Parsons, 1,500 kw.	..	..	100	78	56	33·4
Parsons, 500 kw.	..	..	100	78	56	33·7
Brown-Boveri, 3,000 kw.	..	..	100	77·6	56	33·3
Brush-Parsons, 1,000 kw.	..	120·5	100	79·5	59	38·6
Brush-Parsons, 1,000 kw.	..	119	100	81	61·6	42
Brush-Parsons, 400 kw.	..	120	100	79·7	59·2	39·4
Brush-Parsons, 400 kw.	..	121	100	89	57	37·2
Curtis, 2,000 kw.	..	..	100	77·2	54·2	31·4
Curtis, 500 kw.	149·5	124·5	100	78·2	54·9	32
Zoelly, 500 kw.	..	..	100	77·8	55·7	35·0
De Laval, 200 kw.	..	123	100	76	52	29
De Laval, 20 kw.	..	122	100	82	66	49
Yates & Thom vertical compound Corliss, 1,500 kw.	..	..	100	74	51·5	34·5
Wallsend Slipway Triple-expansion marine type	..	..	100	74	53	32
Alley-McLellan high-speed vertical, 300 kw.	..	..	100	78	56·1	34·2
Reavell, high-speed, 100 kw.	..	130·5	100	79	55·8	..
Belliss & Morcom, 400 kw.	..	..	100	77·7	55·5	33·5
Westinghouse vertical gas engine, 400 kw.	..	..	100	90	84·5	..
Crossley gas engine, 5 kw.	..	..	100	83	62	41

Results of Tests.—The general form of the total water consumption curve is like that of a reciprocating engine with throttle governing; that is to say, it is a straight line. If, however, a by-pass is used at overloads, the consumption curve begins to turn upwards as soon as the by-pass is opened. This is well illustrated by Fig. 164,

which gives the results of tests on a Westinghouse 1,250 kw. turbine with loads up to about 2,000 kw. The Curves I. and II. refer respectively to tests in which saturated steam and steam superheated 77° Fah. was used, the vacuum being 27·1in. in both cases. It will be noticed that the by-pass does not open until about 20 per cent. overload has been reached. Fig. 165 shows the results of tests on a 400 kw. Westinghouse turbine, the results being referred to brake horsepower. A curve showing

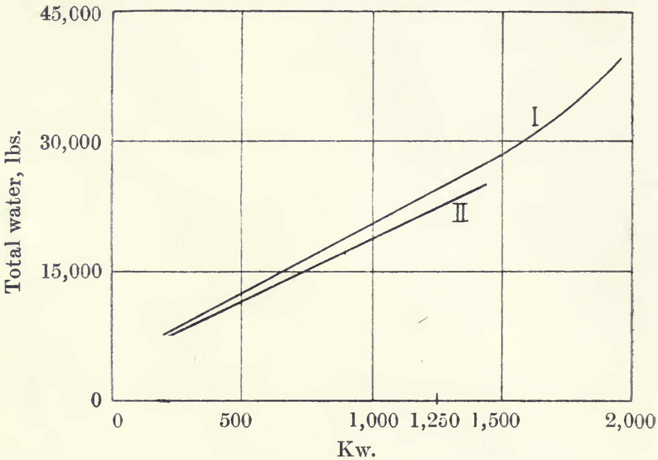


FIG. 164.—WESTINGHOUSE 1,250 KW. TURBINE.
Curve I. = Saturated steam.
Curve II. = Superheat 76° F.

the speed is also added, from which it will be seen that the opening of the by-pass is accompanied by a considerable falling off in the speed, as we should expect.

With regard to the change in the slope of the total-water curve at the point where the by-pass opens it would appear from a consideration of the nature of the change in the conditions inside the turbine cylinder that the curve ought really to be stepped up at this point. Although data as to the actual shape of the curve just at this point are lacking, yet a study of the experimentally determined consumptions before and after the by-pass opens bears out this expectation. (See Figs. 165 and 168.)

In Fig. 166 we have illustrated the results of tests on a 1,250kw. Westinghouse (U.S.A.) turbine. The steam consumptions are referred to brake horse-power and it

will be noticed that the water line droops a little at light loads. The curves also illustrate very clearly the effect of a high vacuum and of superheat.

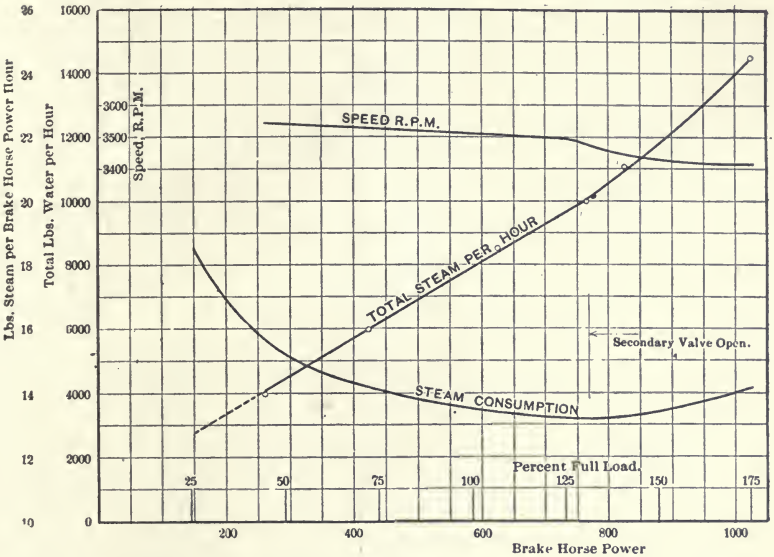


FIG. 165.—RESULTS OF TESTS ON 400 KW. WESTINGHOUSE-PARSONS TURBINE. Steam pressure, 150lbs.; vacuum, 28in.; superheat, 40° F.

Fig. 167 gives the results of tests on a 1,000 kw. Brush-Parsons turbine for Willesden. The two dotted curves are for practically dry saturated steam, and a vacuum of about 27in. The full-line curves give the results with 150° Fah. of superheat, the steam pressure being 167lbs. and 165lbs. per square inch respectively. The gain due to superheating is very marked, 2·27lbs. per kw. at full load per 100° of superheat, or 12·1 per cent. per 100° of superheat. In another Brush turbine the gain was 14·8 per cent. The vacuum with superheat is not given in either case, and it may have influenced the results. The results are good, in particular that with superheated steam. The results are tabulated below :—

Kilowatt load	1,000	750	500	250
Pounds per kilowatt, dry saturated	18·7	20	22·5	29
Pounds per kilowatt 150° superheat	15·5	16·75	19·1	26

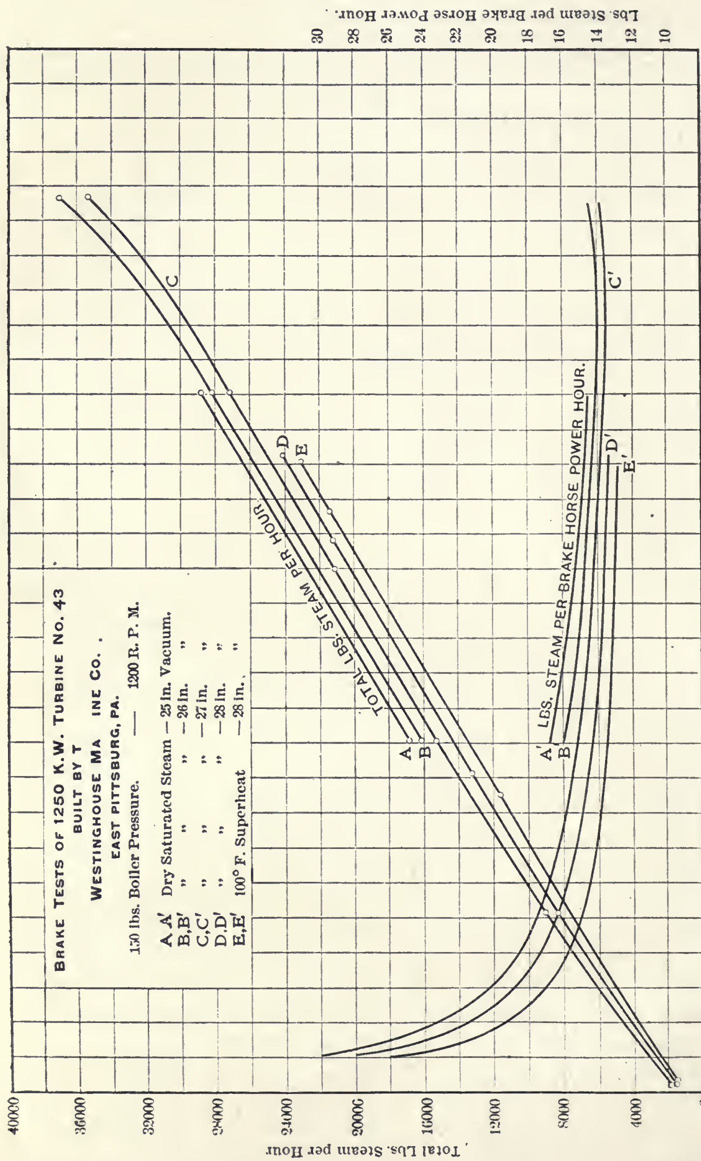


Fig. 166.—Tests on Westinghouse-Parsons Turbine.

In the following table (Table XVII.) are given the results of tests on a 3,500 kw. turbo-alternator at Newcastle. The air and circulating pumps were independently driven. A vacuum augmentor such as is described in Chapter

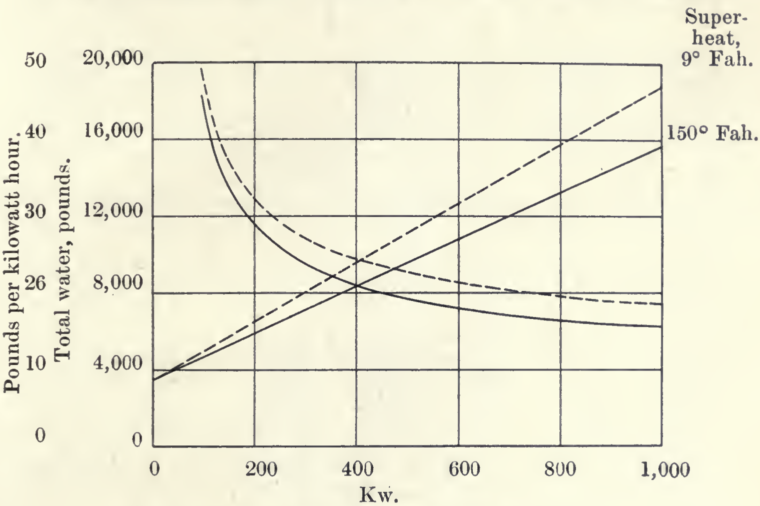


FIG. 167.—TESTS ON BRUSH-PARSONS 1,000-KW. TURBINE.

X. was used to assist in maintaining a good vacuum. The vacuum mentioned in the table is that at the turbine exhaust, not in the condenser.

TABLE XVII.

No.	Kw.	Absolute pressure.	Super-heat ° F.	Speed r.p.m.	Vacuum inches.	Total water.	Lbs. per Kw.	Air Pump Kw.
1	5,801	191·6	132	1,190	28·0	95,500	16·46	13
2	5,685	192·8	151	1,203	28·14	91,100	15·94	15·75
3	4,933	194·4	151	1,200	28·6	78,257	15·87	13·75
4	4,142	199·1	149	1,200	29·0	63,823	15·40	10·4
5	2,108	203·8	152	1,198·2	29·5	34,920	16·56	10·8
6	excited	205·7	93	1,229	29·6	6,062	...	...
7	not excited	206·3	60	1,235·5	29·7	3,920	...	...

Barometer, 30·25 inches.

These results are represented graphically in Fig. 168. The steam consumption curve seems to be straight up to about 25 per cent. overload, and then it begins to rise more steeply. The percentage steam consumptions at

150, 125, 100, 75, 50, and 25 per cent. full load are 154, 124, 100, 77·5, 55·3, and 33 per cent. of the full load consumption.

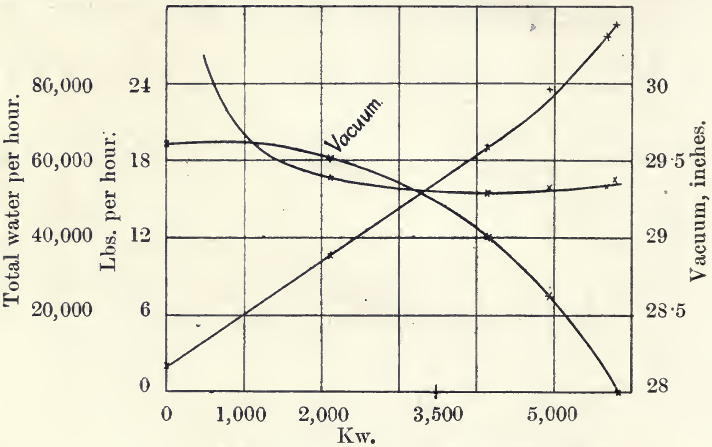


FIG. 168.—3,500 KW. PARSONS TURBINE AT NEWCASTLE.

A few results of tests on turbines by C. A. Parsons and Co. are given below in Table XVIII.

TABLE XVIII.

NEWCASTLE-ON-TYNE ELECTRIC SUPPLY COMPANY, LTD.

1,500 *kw.* 3-*phase Alternator*, 6,000 *Volts*, 40 *Periods*.

Pressure of Steam above Atmosphere at Stop Valve.	Superheat at Stop Valve.	Vacuums in the Turbine Cylinder. Bar=30″.	Revs. per Minute.	Load.	Steam Used.	
Lbs. per Sq. In.	Deg. Fah.	Inches of Mercury.		Kws.	Lbs. per Hour.	Lbs. per Kw. Hour.
196·0	76·0	27·45	1,200	1,442	25,962	18·00
194·0	78·0	26·95	1,199	1,484	26,880	18·10
196·0	85·0	27·45	1,201	1,153	21,456	18·60
197·0	84·0	27·35	1,200	1,015	20,124	19·80
197·0	83·0	27·75	1,200	692	14,874	21·50
196·0	76·9	27·95	1,200	714	15,288	21·40
199·0	77·0	28·35	1,200	360	9,114	25·20
200·0	68·0	28·45	1,200	0	2,948	0
203·0	92·0	26·11	1,210	1,823	32,431	17·70
207·0	66·0	26·46	1,208	1,513	27,582	18·23

NOTE.—The last two tests were made 16 months after the former ones.

SHEFFIELD CORPORATION.

1,500 *kw.* 2-*phase Alternator*, 2,000 *Volts*, 60 *Periods.*

Lbs. per Sq. In.	Deg. Fah.	Inches of Mercury.	Revs. per Minute.	Kws.	Lbs. per Hour.	Lbs. per Kw. Hour.
128·5	125·3	27·52	1,500	1,585	27,855	17·60
131·0	123·6	28·28	1,500	1,071	19,591	18·24
145·0	110·0	28·90	1,500	530	11,453	21·58
148·7	38·4	29·23	1,500	0	2,878	0

HULTON COLLIERY.

300 *kw.* 3-*phase Alternator*, 440 *Volts*, 50 *Periods.*

Lbs. per Sq. In.	Deg. Fah.	Inches of Mercury.	Revs. per Minute.	Kws.	Lbs. per Hour.	Lbs. per Kw. Hour.
161·0	0	0	3,000	297	10,180	34·30
158·0	0	15·33	3,000	297	8,732	29·36
157·0	0	19·33	3,000	305	8,369	27·43
152·0	0	22·33	3,000	303	7,764	25·59
154·0	0	25·33	3,000	303	7,336	24·19
158·0	0	26·58	3,000	303	7,020	23·15

BARNSLEY BRITISH CO-OPERATIVE SOCIETY.

150 *kw.* (*Non-Cond.*) *Continuous Current Generator*, 110 *Volts.*

Lbs. per Sq. In.	Deg. Fah.	Inches of Mercury.	Revs. per Minute.	Kws.	Lbs. per Hour.	Lbs. per Kw. Hour.
145·1	82·3	0	3,440	156	5,703	36·57
151·5	0	0	3,410	154	6,207	40·32
155·6	0	0	3,480	74	3,755	50·76
160·5	0	0	3,500	0	1,349	0

It will be noted that the 150 kw. turbine for Barnsley was run non-condensing, its steam consumption being about 60 per cent. greater than it would have been for a turbine of equal capacity running condensing. This test also shows clearly the effect of superheat on the steam consumption. At practically the same (full) load, superheating the steam 82° reduces the steam consumption 7·6 per cent.

The tests on the 300 kw. machine at Hulton Colliery are instructive as showing the effect that the vacuum has on the steam consumption.

Below are given the results (Table XIX.) of tests on a 3,000 kw. turbo-alternator at Frankfort. Brown, Boveri,

and Co—under license from C. A. Parsons & Co.—are the makers. These results are distinctly good, although it must not be forgotten that the superheat is high.

TABLE XIX.

FRANKFORT.

3,000 *kw.* 3-*phase Alternator*, 3,000 *Volts*, 45 *Periods.*
(*By Messrs. Brown, Boveri, & Co.*)

Lbs. per Sq. Inch.	°Fah.	Inches of Mercury.	R.P.M.	Kws.	Lbs. per hour.	Lbs. per Kw. Hour.
138·5	235·0	27·00	1,350	2,995	44,200	14·74
170·5	187·0	27·50	1,350	2,518	39,300	15·59
142·0	120·0	27·20	1,350	2,600	41,200	15·80
139·0	114·0	27·20	1,360	2,600	41,400	15·90
168·5	184·0	27·90	1,350	1,945	30,800	15·84
146·5	120·0	27·60	1,350	2,000	32,600	16·30
137·0	101·0	27·40	1,350	1,442	25,400	17·60
138·0	52·0	25·70	1,350	644	18,700	29·00
142·0	30·0	28·30	1,350	0	4,700	0
142·0	30·0	28·30	1,350	0	3,560	0

Data respecting the results of tests on Curtis turbines are rather scanty. The following results on a 2,000 kw. machine are given by the makers (General Electric Company of Schenectady, U.S.A.) :—

Kw.	Gauge Pressure.	Superheat Fah.	Vacuum, Ins.	Lbs. per Kw.
2,000	155lbs.	242	28·73	15·3
1,000	160lbs.	242	28·9	16·3
637	150lbs.	215	28·2	20·1

The following results (Table XX.) have recently been obtained (1905) by Messrs. Sargent & Fergusson from a 2,000 kw. Curtis turbine. The last line gives the steam consumption in pounds per kw. hour when a correction of 0·17lbs. has been made for each 10° of superheat :—

TABLE XX.

Kw.	2,023·7	1,066·7	555	—
Superheat, Fah. ...	207	120	204	156
Total water ...	30,400	17,400	10,500	1,510
Lbs. per kw. ..	15·02	16·31	18·09	—
Lbs. per kw. without superheat ...	19·1	18·7	22·1	—

The turbine on which the above tests were made had four stages and (presumably) two sets of moving blades in each stage.

Some tests have recently been conducted on a 500 kw. Curtis turbine in Iowa, U.S.A. The steam was practically dry and saturated. The following results were obtained :—

Load.	Half.	Full Load.	25 per cent. Overload.
Vacuum, inches	28·29	28·01	28·00
Initial pressure	154·6	141·2	147
Lbs. per kw. hour ...	23·8	22·2	21·4

The following table (Table XXI.) gives the results of tests on a 500 kw. Curtis turbine at Cork in 1904 :—

TABLE XXI.

Kw.	Vacuum. Inches.	Superheat, Fah.	Lbs. per Kw.	R.P.M.	Pressure.
128·5	28·8	51	24·9	1,835	155
252	28·6	50	22·64	1,820	155
393·7	27·8	70	20·95	1,822	153
512	26·9	104	20·6	1,820	153
615·8	26·2	124	21·0	1,800	151

Barometer, 30·16in.

The following results (Table XXII.) were obtained on a 1,000 kw. Rateau turbine at the Oerlikon works. The vacuum has been reduced to a 30in. barometer :—

TABLE XXII.

Kw.	1,024	871	659	425	194
Boiler pressure (absolute)	179	181	163	155	186
Pressure at first guide wheel (absolute) ...	116·5	112·2	85·3	57·7	30·4
Vacuum, inches ...	25·05	23·6	26	27·6	27·73
Superheat at first guide wheel, Fah.	10	11	7·3	21	47
Total steam, lbs. ...	22,500	21,500	15,700	10,600	6,200
Lbs. per kw. hour ...	21·98	24·69	23·81	24·91	31·97

TABLE XXIII.

Kw.	470·27	366	280	107·5	—
Absolute stop valve pressure	216	168	170·5	176	180
Superheat at stop valve, Fah.	21	20	15	5	0·5
Superheat at first nozzle, Fah.... ...	21	19	21	33	53
Absolute pressure, first wheel	147	120	95·5	44·7	12·45
Absolute pressure, intermediate receiver...	18·06	14·21	11·41	5·45	1·991
Absolute exhaust pressure	1·85	1·636	1·51	1·294	1·465
Total steam	10,245	8,276	6,560	3,270	980
Lbs. per kw. hour ...	21·78	22·60	23·43	30·42	—
Efficiency dynamo ...	93·4	92·4	92·0	84·0	0·0

The results in the preceding table (Table XXIII.) are taken from Stodola's book on steam turbines, and are some of the results of tests on a 500 h.p. Rateau turbine by Stodola, Wyssling, and Farny.

The air pump was separately driven. The speed was about 2,200 revs. per minute.

A few of Stodola's results obtained from a Zoelly turbine are given below (500 h.p.) :—

TABLE XXIV.

Kw.	Absolute Pressure.	Superheat, Fah.	Vacuum.	Total Steam.	Lbs. per Kw.
364	164	6·3	—	7,700	21·77
388	164	7·0	28·62	7,810	21·48
335	160	3·8	28·59	7,440	22·20
241	162	4·0	28·47	5,800	24·07
183	161	4·0	28·41	4,700	25·70
81	162	3·1	28·70	2,680	33·07
392·5	188	103	28·80	7,470	19·03
390·4	193	121·5	28·74	7,170	18·38
391·2	165·5	76·5	28·74	7,740	19·80

The steam consumption at no-load (dynamo excited) was 1,025lbs. per hour, the superheat being 3°. The steam consumption at light loads seems to be rather high relatively to the full-load consumption.

TABLE XXV.

Date of Test	26-2-05	27-2-05	27-2-05	27-2-05	27-2-05
Useful output, kilowatts	132·19	208·21	291·52	391·13	463·22
,, B.H.P...	229·34	345·11	459·28	597·27	697·91
Revolutions per minute	3,061	3,050	3,040	3,030	3,020
Pressure at inlet, lbs. per sq. in.	126·83	124·63	125·07	124·92	125·36
Temperature ° Fah.	339	339	339	339	339
Pressure before 1st guide, lbs. per sq. in.........	39·83	55·85	73·48	95·97	111·84
Vacuum, inches	28·5	28·3	28·1	27·8	27·5
Steam consumption per hour, lbs.............	4,123	5,472	7,143	9,162	10,624
Steam consumption per kw. hour, lbs.	31·19	26·28	24·5	23·43	22·9
Steam consumption per B.H.P. hour, lbs.	17·98	15·86	15·55	15·34	15·22
Thermodynamic efficiency per cent.	45·4	51·6	53·4	55·03	56·4

The above figures give the results of tests on a recent Zoelly turbine of 550 b.h.p. The turbine was direct connected to two continuous-current dynamos at Mülhausen. The steadiness of the speed was such that it did not vary by 3 per cent., when the full load was applied or removed.

The table on page 264 (Table XXVI.) gives the results of tests on different sizes of De Laval turbines.

Tests on Low-pressure Turbines.—As has been repeatedly pointed out, a turbine can make better use of a vacuum than a reciprocator, but not quite such good use of that portion of the expansion above, approximately, atmospheric pressure. For this reason, combinations of non-condensing reciprocators with low-pressure condensing turbines have been suggested, and a few actually installed.

At a mine at Bruay, in France, a low-pressure turbine has been used to assist the reciprocating engines. The turbine is of about 300 h.p. and receives steam at or below atmospheric pressure. Below (Table XXVII.) are

TABLE XXVI.
Results of Tests with De Laval Steam Turbines at Different Loads.

Turbine Machine.	Pressure of Admission Steam. Lbs. per sq. inch.	Vacuum, Inches of Mercury.	No. of Nozzles open.	Electrical H.P.	Lbs. of Steam per Electrical H.P. per hour.	Remarks.
50 H.P. Turbine	113·8	26·3	6	49·4	24·6	Work for
Dynamo.	113·8	26·3	5	40·2	25·2	condens-
The test made	93·9	26·9	4	25·0	27·9	ing
in April, 1895.	74·0	27·5	3	12·7	32·5	included.
100 H.P. Tur-	103·7	25·8	5	92·7	22·6	Work for
bine Dynamo.	103·8	26·4	3	55·6	22·7	condens-
The test made	107·4	26·8	2	35·0	24·7	ing not
in June, 1897.	106·7	27·9	1	15·5	27·8	included.
				Brake H.P.	Lbs. of Steam per Brake H.P. per hour.	
150 H.P. Tur-	113·8	26·4	7	163·0	17·6	Work for
bine Motor.	116·9	25·9	6	138·4	18·2	conden-
The trial made	113·8	26·2	5	114·5	17·9	sing not
in November,	114·3	26·5	4	88·3	18·7	included.
1897.	112·4	27·0	3	64·1	19·0	
	116·2	25·7	2	37·5	22·3	
300 H.P. Tur-	192·7	27·3	7	303·6	14·1	Work for
bine Motor.	196·3	27·6	6	255·5	14·7	conden-
The test made	196·3	27·6	5	216·9	14·4	sing not
in December,	196·3	27·6	4	172·6	14·5	included.
1899.	190·6	27·8	3	121·6	14·9	
	196·3	28·1	2	74·2	17·2	
	213·3	28·5	1	31·5	21·6	
300 H.P. Tur-	126·6	26·98	8	337·45	15·68	Work for
bine Motor.	126·4	26·99	7	293·7	15·76	conden-
The test made	125·0	27·24	6	249·1	15·92	sing not
in June, 1900.	125·0	27·62	4	162·7	16·25	included.
	125·0	27·91	3	118·9	16·70	
	125·0	28·16	2	73·5	18·00	
	125·0	28·25	1	30·4	21·77	

In the tests with the 300 h.p. turbine, the steam was superheated 60° Fah. in the former case and about 20° Fah. in the latter.

When dealing with the subject of governing, we gave some figures illustrating the consumption of steam for De Laval turbines with nozzle and with throttle governing.*

* Chapter VI.

the results of some tests on this turbine when running at from 1,300 to 1,800 revs. per minute. The efficiency given in the table refers to the ratio of the power at the dynamo brushes to the theoretical work in the steam.

TABLE XXVII.

Kw.	Lbs. absolute.		Temperature, Fah.	Superheat, Fah.	Lbs. per H.P.		Efficiency.
	Admission.	Condenser.			Theoretical.	Actual.	
70·3	5·42	1·25	231·8	66	25·1	51·1	0·492
140·9	9·37	1·82	275	85	22·3	42·1	0·530
202·9	12·83	2·32	278·6	73	21·1	39·7	0·531
232·5	14·70	2·79	296·6	84	21·9	39·5	0·556
169·5	12·02	2·13	269·6	67	21·9	45·2	0·485
186·0	12·02	2·22	269·6	67	22·2	40·6	0·547
198·0	12·02	2·22	271·4	69	22·2	38·1	0·581
232·5	14·37	2·52	275·0	64	21·3	39·8	0·534
247·0	14·37	2·62	275·0	64	21·7	37·4	0·580

A low-pressure Curtis turbine* has recently been installed in Philadelphia. It takes the exhaust steam from reciprocating engines at a pressure of 15lbs. or 16lbs. absolute, and exhausts into a condenser with an average vacuum of 28in. The turbine has no governor, but takes all the steam the engines will supply. The turbine is really a Rateau turbine in principle, as the number of rows of moving blades per stage is only one. The turbine is said to increase the output over and above that obtained by the non-condensing reciprocators by about 66 per cent. for the same steam consumption. With dry saturated steam at atmospheric pressure, and 1lb. back pressure—28in. vacuum—the guaranteed consumption is 36lbs. at full load (800 kw.) and 40lbs. at half load. With 2lbs. back pressure—26in. vacuum—the guarantee is 45lbs. and 50lbs. respectively.

Since the steam as it exhausts from reciprocating engines is very seldom superheated and is often wet, it is good practice to insert a steam separator in front of the turbine inlet. Wet steam must be avoided as far as possible.

* "Electrical World and Engineer," December, 1905.

A field of the same kind which does not yet seem to have been tested is the generation of low-pressure steam by means of the exhaust from gas engines, the steam to be used in a low-pressure steam turbine.

Tests of Turbo Pumps.—Owing to the high speed of rotation, steam turbines are suitable for direct connection to high-lift centrifugal pumps. Efficiencies of over 70 per cent. and even as high as 80 per cent. are claimed for such pumps, but these figures seem too high. The results of tests on a Rateau turbo pump in Bohemia are given in Table XXVIII.

TABLE XXVIII.

Admission, lbs. per square inch	82·5	92	94·6	97·5	104
Vacuum, lbs. per square inch	2·47	2·47	2·47	2·47	2·47
Useful water horse-power	89	107·8	116·7	123·6	139·4
Theoretical lbs. per h.p. hour	11	10·7	10·62	10·5	10
Actual lbs. per h.p. hour	34·6	32·1	30·3	29·5	27·8
Net efficiency	0·315	0·335	0·350	0·355	0·360

The overall efficiency is thus about 35 per cent., which is about the same as that obtained by Prof. Goodman in some tests on a Parsons turbo pump for New South Wales, the results of which are given in the table on page 267 (Table XXIX).

Results of Tests on Reciprocating Engines.—A few representative results obtained with reciprocating engines are given in Table XXX. (page 269) for comparison with the results obtained from turbines. The reader is warned, however, against attempting to press the comparison too closely. Comparisons between individual turbines and reciprocators are seldom of much use; indeed, one has to try to obtain a mental average of the results for both classes of prime mover when allowances have been made for the conditions under which they operate.

None of the above tests have been selected as showing specially good results, but rather as representing

TABLE XXIX.

No. of test	1	2	3	4
Date	Jan. 27, 1903	Feb. 3, 1903.	...	...
Title of test	Contract.	Contract.	Overload.	Superheat.
Duration of test in hours	2	1·54	1·2	1·55
Steam pressure in lbs. per sq. in.	57	57	84	55
Vacuum in condenser in inches of mercury	27·46	27·82	27·62	27·80
Barometer, inches of mercury	29·65	29·89	29·89	29·89
Condition of steam	Dry and saturated.	Saturated	Saturated	95° F. Superheat.
Weight of steam used per hour in lbs.	7,039	7,208	9,398	6,667
Weight of steam per water h.p. hour in lbs.	27·93	30·67	28·83	27·89
Speed, revs. per min.	3,300	3,330	3,710	3,340
Mean delivery pressure gauge readings in lbs. per square inch—				
1st pump	97	102	130	103
2nd pump	206	208	263	210
3rd pump	326	318	393	323
Height of suction in feet	11	11	11	11
Total lift in feet	762	744	917	756
Millions of gallons of water pumped per day—				
Venturimeter	1·573	1·499	1·689	1·503
Orifice	1·623	1·513	1·723	1,555
Water h.p.	252	235	326	239
Gross h.p. that would be obtained from a perfect steam engine working on the Rankine cycle between the given limits of temperature	720	759	1,049	733
$\frac{\text{Water h.p.}}{\text{Gross h.p.}} \times 100$	35·0	31·0	31·1	32·6

good standard practice. The tests on the first three sets in the table were made at the Manchester Corporation Electricity Works. Those on the two Pollit and Wigzell engines were official tests carried out at the Halifax and Southampton Electricity Works respectively. The test on the Allis-Chalmers engine was carried out in the power-house of the Interborough Rapid Transit Company of New York. This engine has four cylinders, two vertical and two horizontal, being in fact a double-compound engine. The total friction losses of the engine and its generator were determined by rotating the engine by means of a motor. The losses only amounted to 417·3 kw., or 7·6 per cent. of the rated output.

The general agreement among engineers is that there is nothing to choose between steam turbines and steam reciprocators, so far as steam consumption goes, provided the turbine vacuum be good, say, 27·5in. or more.

With a vacuum of about 25in. or 26in., there is no doubt but that the turbine is inferior to the reciprocator. For small powers the reciprocating engine is also distinctly superior to the turbine, and of the different types of turbine the reaction turbine is the most affected by smallness of size. Even for large powers the all-day economy of a reciprocating engine is probably in general somewhat superior to that of the turbine, particularly so where the engine is governed by varying the cut-off. The above remarks refer, of course, to steam consumption. Where the turbine frequently gains is by a reduction in fixed charges owing to reduced capital outlay. Unfortunately, in a too often blind endeavour to make a good showing on test, the turbine installation is overburdened with a too expensive condensing equipment. We shall, however, refer to this subject again later on.

So far as can be ascertained there is some saving in oil with turbines, particularly if the reciprocators are of the open type. On the other hand, the turbine condenser requires more water than that of the reciprocator, which in many cases is a matter of much more consequence than the oil. There are other things to consider besides operating costs, however, and these will be dealt with later.

Returning to the consideration of the relative steam economies of the turbine and reciprocating engine, we

TABLE XXX.

Engine.	KW.	Gauge Pressure.	Vacuum Inches.	Superheat Fah.	Lbs. per Kw.	Revs. per minute.
Musgrave-Westinghouse vertical compound 1,800 kw.	1,836 1,503	155 155	27·5 27·5	... 90	19·62 17·2	74·1 75
Yates & Thom 1,500 kw. vertical compound condensing.	1,500 1,225 750 375	160	25·5	... ...	20·4 20·2 21·47 28·0	94 ... ...
Similar set, non-condensing.	1,500 1,225 750	160	... ...	... ...	27·0 29·1 34·5	94 ... ...
Alley-McLellan, 300 kw.	330 300 225 150 75	... ...	... ...	... ...	21 21·2 22·1 23·8 29	300 ... ...
McIntosh & Seymour vertical 1,500 kw.	1,500 750	158 ...	24·2 26	92·3 ...	16·5 15·75	... ...
Pollit & Wigzel, 750 kw.	723·6	146·6	26·4	180	17·9	90
Pollit & Wigzel, 500 kw.	532	161	25·8	154	18·7	100
Allis-Chalmers, 5,500 kw.	5496·5	175	26·0	? 0	16·0 17·34*	75
Belliss & Morcom, 400 kw.	360	165	25·4	200	16·1	340
Belliss & Morcom, 1,500 kw.	1,436	183	25·6	64.4	19·15	200

* Same trial as preceding figures, equivalent dry saturated steam.

have already shown that the steam economy is practically the same for both. Starting from this fact, we will examine a little more closely into their relative performances. In Fig. 169 we have an entropy diagram. The figures *E F G H I J* and *K L M N P Q* are the high and low pressure diagrams of a compound reciprocating engine working between the pressures *B C* and *A D*. The effects of the steam shut up in the clearance space have been eliminated by an accurate method due to Wilson and Noble.* The sum of the areas of these two figures gives the total amount of work obtained from each pound of steam actually passing through the engine. The lines *E F G* and *K L M* are the admission lines for the two cylinders, *G H* and *M N* are their expansion lines,

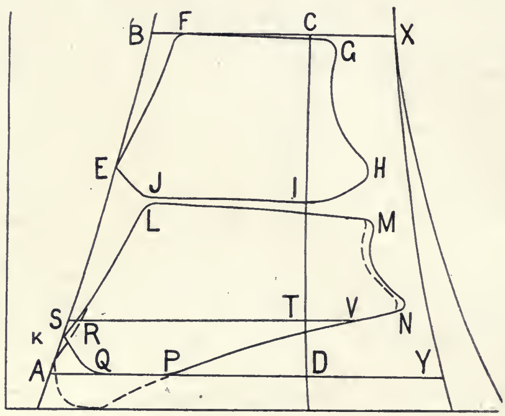


FIG. 169.—RELATIVE EFFICIENCIES OF TURBINES AND RECIPROCATORS AT VARIOUS POINTS DURING EXPANSION.

whilst *H I* and *N P* are the curves of release at approximately constant volume. The slight upward tendency of the curves at *J E* and *Q K* is due to the throttling just previous to compression.

Suppose that the turbine converts 65 per cent. of the theoretically available work between the limits of pressure into "indicated" work.† Then, by methods discussed elsewhere in this book, we can draw the expan-

* "On the Construction of Entropy Diagrams from Steam Engine Indicator Diagrams." Memoirs and Proceedings of the Manchester Literary and Philosophical Society, vol. 45, part IV.

† It would probably be more accurate to assume the efficiency of the turbine as greater at low than at high pressures by about 5 per cent.

sion line $X\ Y$—we have assumed that there is no initial condensation—from which we can draw the curve $C\ D$ such that at all points on it the distance from the curve to the line $A\ B$—measured horizontally—is 65 per cent. of the distance between $A\ B$ and $X\ Y$. Then the area $A\ B\ C\ D$ represents the work obtained by the turbine from each pound of steam, and is therefore equal to the two areas $E\ F\ G\ H\ I\ J$ and $K\ L\ M\ N\ P\ Q$. From the shape of the reciprocator diagram at low pressures we

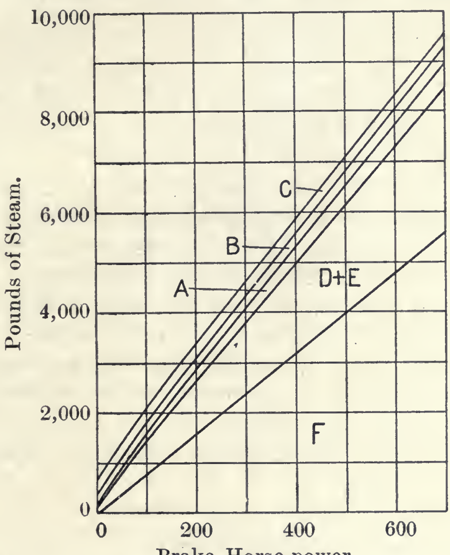


FIG. 170.—ANALYSIS OF TEST ON 400 KW. WESTINGHOUSE-PARSONS TURBINE.

see that the reciprocator must be the more economical of the two down to about release pressure. At any pressure if we draw the horizontal line $S\ R\ T\ V$ then $R\ V$ and $S\ T$ are respectively proportional to the works obtained from the reciprocator and turbine at that pressure.

We can now ascertain with fair accuracy the relative value of high vacua to reciprocators and turbines. The release curve of the reciprocator will always lie along $N\ P$, because the volume of the low-pressure cylinder at release is in no way affected by the vacuum. The rest of the diagram will be slightly altered by a change in the vacuum.

The higher the vacuum the greater the temperature variation in the cylinder between admission and exhaust, and hence the greater the initial condensation, so that the shape of the diagram will now be somewhat as shown dotted in the figure. Still, it is nearly correct to say that the extra indicated work obtained from the reciprocator is to that obtained from the turbine as $R\ V$ is to $S\ T$ for a small change in the vacuum at the pressure $S\ T$. It is clear, then, that it is only below a pressure of about 7lbs. to 9lbs. absolute that the turbine becomes more efficient than the reciprocator.

Analysis of Test.—Whilst it is not possible to make a complete detailed analysis of all the losses which take place in a steam turbine, yet it is possible to make a partial analysis which is of some interest.

The theoretically available work in the steam used by the turbine may be subdivided as follows :—

W = total theoretical work in steam between admission and condenser pressures.
A = loss to exhaust, mainly kinetic energy.
B = radiation and leakage from turbine.
C = work done in overcoming bearing friction.
D = work done in overcoming the resistance of the steam to rotation.
E = loss by wiredrawing (internal leakage) and friction.
F = work available at the shaft.

Then $W = A+B+C+D+E+F$.

For convenience, we will express all the above quantities in terms of pounds of steam. Thus, a 5 per cent. loss will be represented by 5 per cent. of the steam used.

The loss to exhaust A, the steam resistance D, and the losses E—which are equivalent to a throttling effect—are roughly proportional to the steam passing through the turbine. The radiation and leakage loss will also increase somewhat with the load, the bearing friction, however, being practically constant. With constant initial and condenser pressures, F will be strictly proportional to the brake horse-power. It will be seen from this analysis that the steam used at no load is not all spent in overcoming the frictional horse-power of the turbine. Some of it is accounted for by radiation and leakage, some by

the loss to the exhaust, some by the "throttling" effect E due to internal leakage, eddies, and skin friction in a direction perpendicular to the direction of rotation.

In Fig. 170 an attempt has been made to analyse the results of some tests on a 400 kw. Westinghouse turbine using dry saturated steam at 165lbs. absolute, and exhausting into a vacuum of 1lb. absolute. The theoretical work in each pound of steam is (W) 318 B.Th.U., so that at 700 b.h.p. (full load) the steam equivalent to the brake horse-power is (F) 5,600lbs. The total steam consumption at full load is 9,550lbs., and 650lbs. at no load. It will be noted that the curve is not quite straight, but sinks slightly at light loads. From the small no-load consumption—only 6·8 per cent. of the full-load consumption—it is clear that the bearing friction, radiation, and leakage are only small. In our analysis we have assumed the bearing friction C constant at 2 per cent. of the full load, or equal to 190lbs. Radiation and leakage B we have taken as increasing from 180lbs. to 320lbs. The loss to exhaust we have taken as increasing from 50lbs. to 500lbs. D and E have been grouped together, because it is difficult to separate them, and because the effect of both is equivalent to a throttling effect. In this case D and E total up to 2,940lbs at full load, or 31 per cent. of the total consumption The net thermal efficiency of the turbine at full load is 58·7 per cent. Where the vacuum or initial pressure varies with the load it would perhaps be advisable to draw the diagram on a heat, instead of steam, basis. The diagram brings out clearly the relative importance of the losses. Clearly, the greatest losses are frictional, eddy, and leakage losses. Hence the necessity for great care in the arrangement and formation of the blades.

In Fig. 171 the various components of the initial work in the steam are represented as percentages.

Owing to the uncertainty which there is as to the exact no-load water consumption the precise percentage losses as given in Fig. 171 at or near no-load are not very reliable.

The percentage external leakage in the above test is less than that usually obtaining in practice. This leakage takes place chiefly over the ends of the balance

pistons, and is seldom less than 5 or 7 per cent. of the total steam passing. It depends, of course, on the axial clearances at the balance piston rings.

General Deductions.—Most of the tests described so far have been simply for the purpose of determining the steam consumption under normal conditions. The results, however, teach us something more than the particular figures of steam consumption.

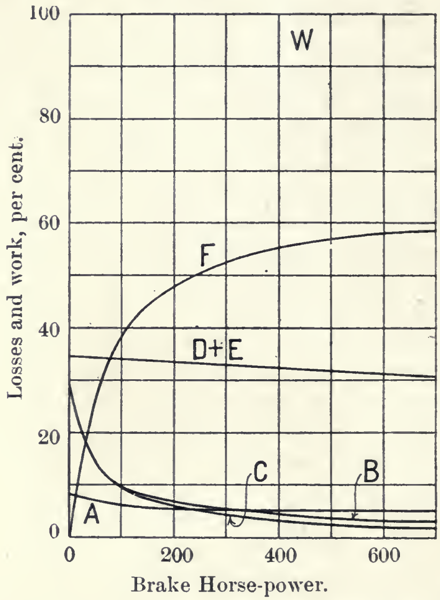


Fig. 171.—Analysis of Losses in Westinghouse-Parsons 400 kw. Turbine.

W—Theoretical work.
A—Loss to exhaust.
B—Radiation and external leakage.
C—Bearing friction.
D—Disc friction.
E—Internal hydraulic losses.
F—Work done on shaft.

In the first place, it will be noticed that whereas the vacuum has a very marked effect on the consumption, the effect of the initial pressure is by no means obvious and is not very great for pressures above about 130lbs. or 150lbs. per square inch. This is, of course, what we might expect from theory. Thus at 160lbs. absolute the available work with a 27in. vacuum is 297 B.Th.U.,

whilst at 180lbs. the available work is only 305, an increase of about 2·7 per cent., with an increase in the initial pressure of 20lbs. per square inch. On the other hand, an increase in the vacuum to 28in. increases the available work by 6·3 per cent. (See Table in Appendix.)

Again, although the number of stages should theoretically vary directly as the available work, yet the results of actual tests show that the number of stages can be varied very considerably without having any marked effect on the efficiency. Thus below are given the net thermal efficiencies of the Hulton Colliery turbine at various vacua and approximately constant load. Although a condensing turbine, the maximum efficiency is obtained when running non-condensing, and the efficiency falls off as the vacuum increases.

Vacuum, Inches.	Pounds per Kilowatt-hour.		Efficiency.
	Theoretical.	Actual.	
0	19·05	34·03	·555
15·33	15·46	29·36	·527
19·33	14·45	27·43	·527
22·33	13·32	25·59	·521
25·33	12·16	24·19	·503
26·68	11·55	23·15	·499

The best number of stages should be inversely proportional to the theoretical pounds per kilowatt-hour. Yet we see that with a fixed number of stages instead of a *decrease* of 39 per cent. there is an increase in the efficiency of 5·6 per cent. This result is doubtless not wholly due to the ratio of the number of stages to the theoretical number required, but it will serve to show that the number of stages can be varied within quite wide limits. The results of other tests emphasize the same point.

Chapter XII.

MARINE TURBINES: COMPRESSORS: GAS TURBINES.

Marine Turbines.—The steam turbine as a propelling agent for ships, possesses, at first sight, overwhelming advantages over the ordinary reciprocating engine. It has not to convert a reciprocating motion into a rotary one, it is a perfectly balanced machine, it is as economical as the reciprocator, in land practice it is lighter and smaller than the corresponding reciprocator, and requires rather less attention. Unfortunately, all these advantages are not entirely realised, and there are some disadvantages.

The turbine is naturally a very high speed machine; too high, in fact, for marine work, so that its speed must be reduced. In the process of this speed reduction, the turbine becomes bulkier and more expensive. Several of the recently-installed marine turbines have cost at least as much as the reciprocating engines of their sister ships. It is not merely that the rotational speed must be reduced, but the peripheral speed of the blades as well, in order to reduce the otherwise impossibly large diameters of the turbine rotors and also to reduce the centrifugal stresses in the rotor so that the danger due to the propeller racing may not be serious.*

For instance, suppose a peripheral speed of 350ft. per second were attempted for the blades. The turbine rotors of the "Victorian," running at 260 revs. per minute, would have a mean diameter of 26ft.; those of the "Carmania" at 180 revs. per minute a diameter of 37ft.; and those of the new 25-knot Cunarders at 140 revs. per minute a diameter of no less than 48ft.

Then, too, a high peripheral speed with a low rotational speed will in many cases make the blade heights impossibly

* For table of speeds see page 109.

small, and the clearance leakage over the blade tips very great. At a given number of revolutions per minute

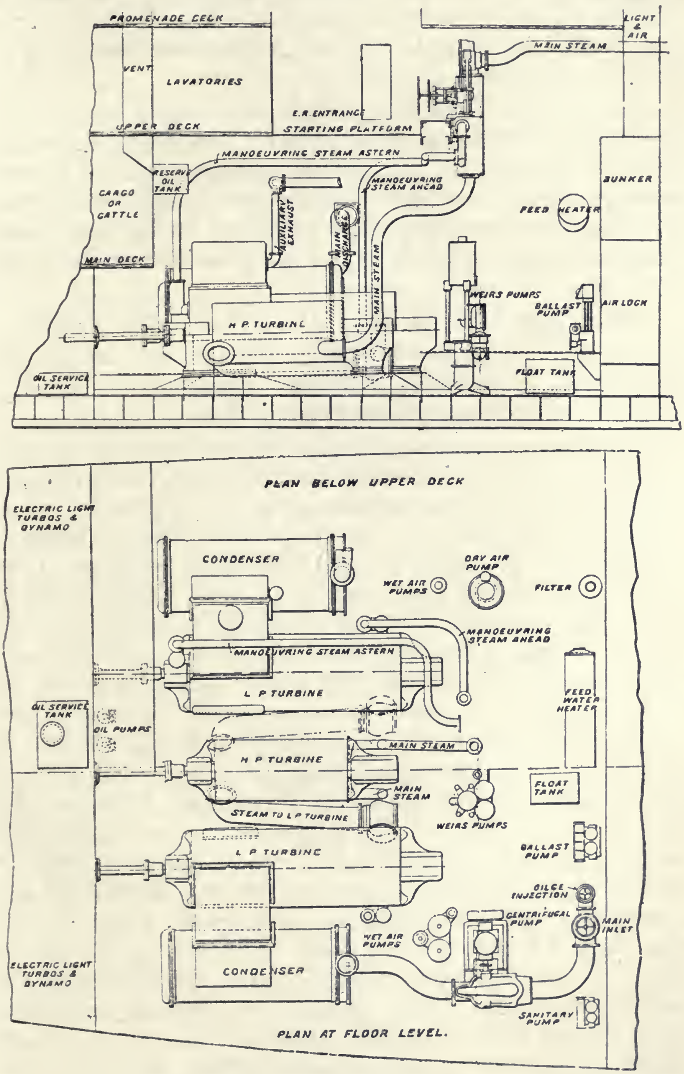


FIG. 172.—TURBINE ROOM OF TURBINE-DRIVEN MIDLAND RAILWAY BOATS.

with fixed blade shapes, the radial distance between the rotor and casing will vary inversely as the square of the

peripheral speed. It is largely for this reason that the application of reaction turbines in ships is limited to vessels of high speed—allowing a fairly high rotational speed—or very large size in which the steam consumption is large and allows of fairly long blades.

This reduction in the peripheral velocity is a serious drawback. In the turbines of the "Victorian" there are over a million and a half blades, besides which, low peripheral velocities are not conducive to high efficiencies. Still, the steam turbine does not seem to lag behind the reciprocating engine for high-speed marine work so far as economy of coal is concerned.

As regards the difficulty of applying the turbine to low-speed, low-powered ships, perhaps we cannot do better than quote Mr. Parsons. Speaking before the Institute of Marine Engineers he said :—

"With the evidence at present before them, he thought they were safe in predicting that the steam turbine would soon entirely supersede the reciprocating engine in vessels of 16 knots sea speed and upwards, and of over 5,000 h.p. It would also probably be used for vessels of speeds down to 13 knots, and of 2,000 tons and upwards, and possibly, also, in even slower vessels in course of time. At the present it might be said that the above most suitable field comprised about one-fifth of the total steam tonnage of the world. It must be remembered, however, that the speed of ships tended to increase, and the turbines to improve ; thus the class of vessels for which the turbine was suitable would increase. It seemed probable also that a combination of the reciprocating engine and the turbine would be found the best machinery for vessels of the tramp class in the immediate future. That field was a very large and important one, and deserved more attention than it had as yet received. The case might be put thus : In a slow vessel, say of 10-knot speed, the revolutions had to be low, because a certain disc area of propeller and a certain number of square feet of blade area were necessary in order to avoid too great a slip ratio and consequent loss of propeller efficiency. It had been proved by experiment that the pitch ratio could not be reduced much below ·8 without commencing to incur excessive loss from skin friction of the blades.

So they had no alternative but to accept the highest revolutions possible under the circumstances, which, in a 10-knot vessel, was a low figure. To obtain a reasonable economy from a turbine a certain surface speed of the turbine blades was essential, as well as a certain number of rows of blades. If the revolutions were low, the diameter and also the number of rows must be increased in order to maintain the economy. The result was that for 10 knots the diameter and number of rows became inordinately great, the weight and cost became excessive, and further, the efficiency of the turbine was somewhat

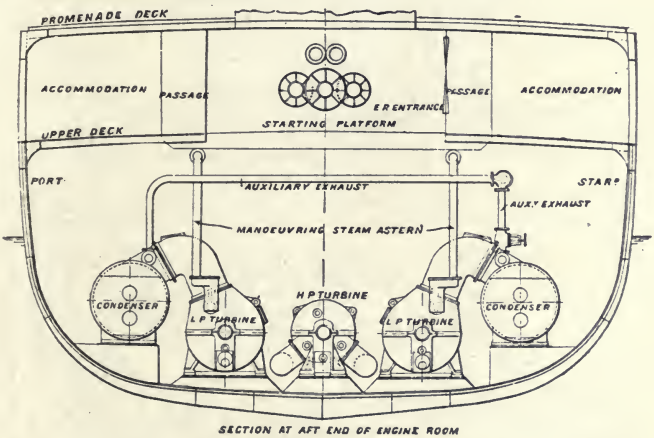


FIG. 173.—CROSS SECTION OF TURBINE ROOM OF MIDLAND RAILWAY BOATS

impaired by such extravagant dimensions in proportion to the power realised. They were therefore blocked on those lines."

For very small vessels, such as steam launches, a high-speed turbine (preferably of the impulse type, if the power is below 200 h.p.) may be used and geared down after the manner of the De Laval turbine. The advantage of the impulse turbine for small powers lies in the fact that it is possible to use partial peripheral admission and reasonable blade heights at fair peripheral speeds. With a reaction turbine partial peripheral admission is not desirable, and hence either the blade heights will be small—due to the large diameter necessary to obtain a sufficiently high peripheral speed—or the diameter and peripheral speed will be low.

Where gearing is inadmissible the impulse turbine, for the reasons outlined above, is superior to the reaction turbine for small powers and low speeds.

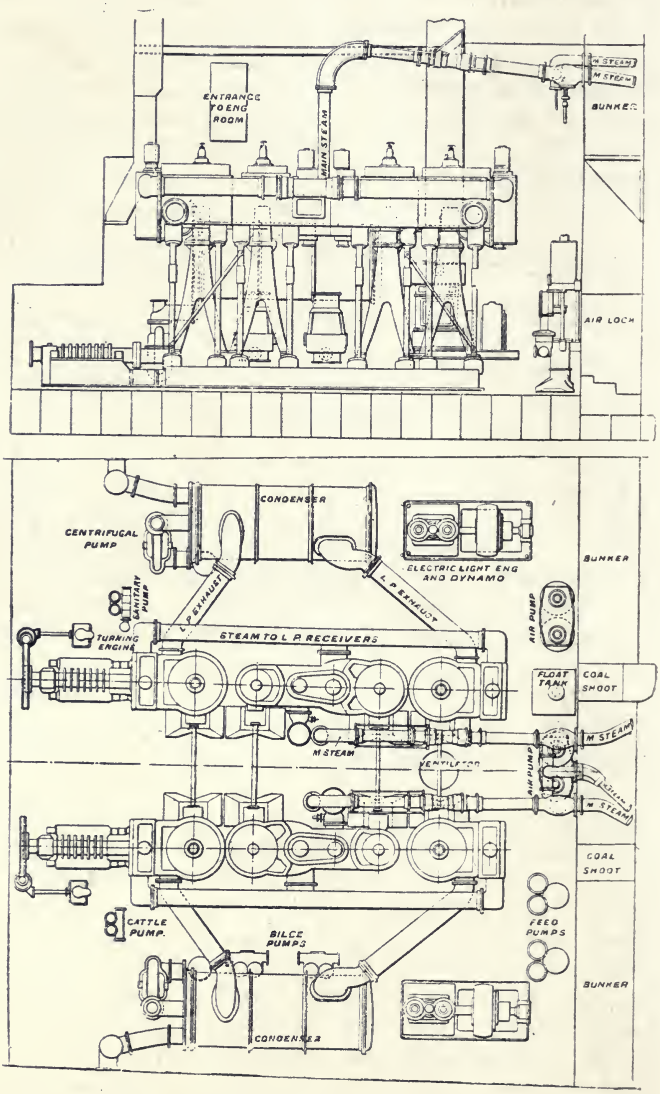


FIG. 174.—ENGINE ROOMS OF MIDLAND RAILWAY BOATS.

Since for small powers one of the determining factors is the blade height, it is frequently desirable in such cases to employ very acute blade angles so as to reduce the axial velocity of the steam and increase the blade height. Provided the inlet and outlet angles have the correct relative values, the work done per stage, and hence the number of stages, will not be affected by this decrease in the axial velocity of the steam. For reasons that were discussed when dealing with blades, this may, however, somewhat increase the friction loss, but will largely reduce the leakage over the blade tips.

Where, as in slow-speed ships, the highest practicable blade speed is small, it is desirable to use acute blade

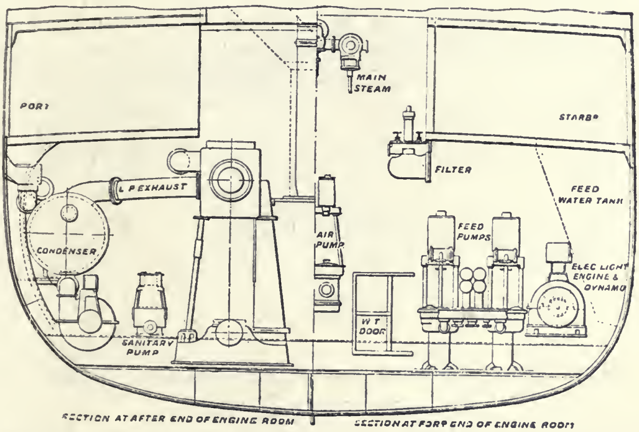


FIG. 175.—CROSS SECTION OF ENGINE ROOM OF MIDLAND RAILWAY BOATS.

angles, so as to increase the work done per stage and hence decrease the number of stages. For the same reason, viz., to reduce the cost of the turbines, it may be desirable to cut down the number of stages below the calculated number. This will not by any means cause a proportionate decrease in the steam economy, because, owing to the steeper pressure gradient along the turbine, the steam velocities and hence the work per stage will be increased and the friction losses will be reduced because of the reduced area exposed to the rubbing action of the steam.

Since the conditions vary so much, it is impossible in the present state of the science and art of turbine propulsion to lay down a set of rigid empirical rules. Each case must be considered on its merits. It is to be hoped that the designer will not fall into the error of imagining that all marine turbines should be alike as to blade angles and general construction; differing only in respect of the number of stages and peripheral speeds, the former being inversely proportional to the square of the latter. As we have just pointed out, departures from standard proportions may increase the commercial efficiency of

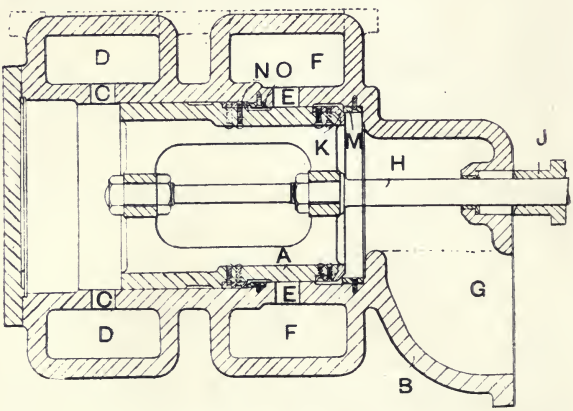


Fig. 176.—Parsons' Reversing Piston Valve for Steam Turbines.

the turbine. In turbine work, as in most other things, of two evils choose the lesser.

As regards the relative economy of marine reciprocators and turbines we are not as yet justified in dogmatising. Judging by the results obtained in land practice, we should hardly expect any gain; still, some of the turbine boats built to date do seem to be more efficient on trial than their sister reciprocating-driven ships. Some interesting results bearing on this point were obtained with the cross-channel boats, "Antrim," "Donegal," "Londonderry," and "Manxman." The first two named have reciprocating engines, and the latter two have turbines. The principal dimensions of the three former vessels are as follows: Length on the water

line, 330ft.; moulded breadth, 42ft.; moulded depth, 25ft. 6in. The "Manxman" is similar in form, except

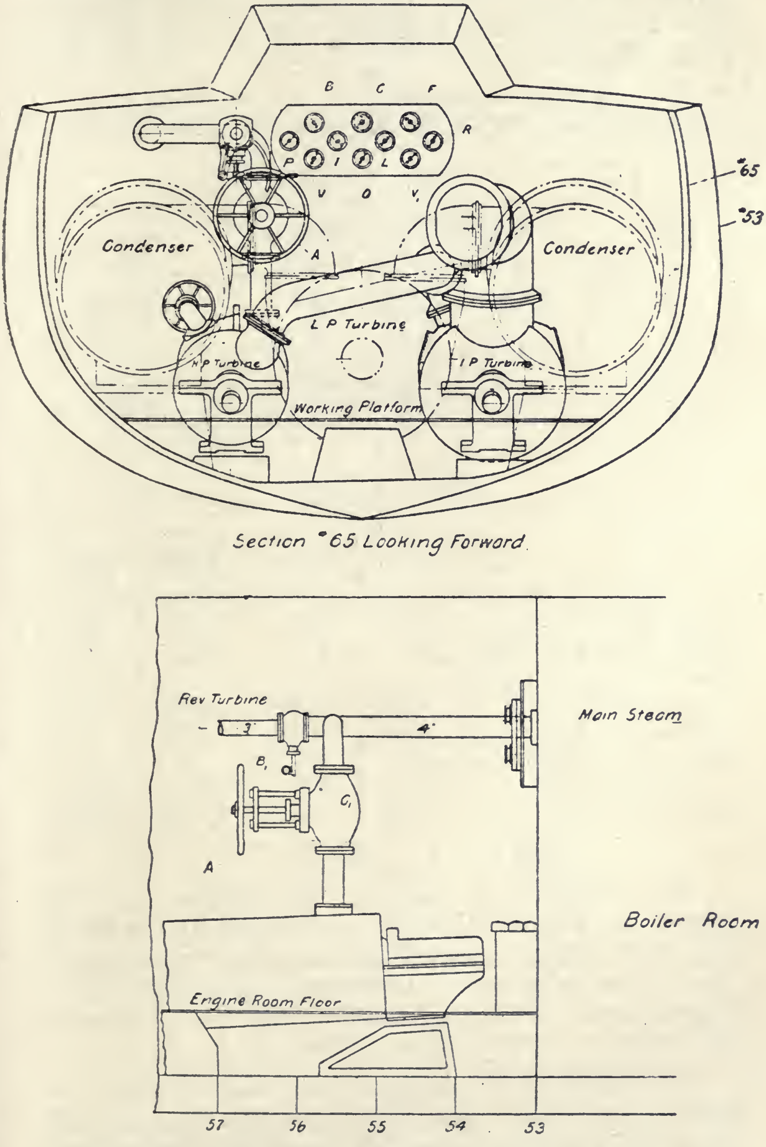


FIG. 177.—ARRANGEMENT OF TURBINES ON THE "TARANTULA."

that her breadth is 43ft. The boilers in all four vessels are alike as to size, the "Londonderry" having a steam pressure of 150lbs. per square inch, and the others 200lbs. per square inch. The "Antrim" and "Donegal" have 4-cylinder triple-expansion engines each driving a 3-bladed propeller.

The water consumptions of the different ships at various speeds on trial runs are given in Table XXXI.

TABLE XXXI.

Knots. Speed.	"Antrim" and "Donegal."	"Londonderry."	"Manxman."
14	4,500	4,500	4,500
17	6,700	6,100	5,800
20	9,700	8,900	8,300
22	...	13,600	12,500
23	...	..	17,300

TABLE XXXII.

	"Antrim."	"Donegal."	"Londonderry."	"Manxman."
Number of passages...	77	81	90	68
Coal per passage, tons	36·7	37·2	36·1	39·6
Hours at full speed per passage	5·78	6·07	5·81	5·35
Hours per passage ...	6·77	6·92	6·78	6·73
Percentage of hours at full speed	85·5	87·7	85·7	79·5
Coal per hour	5·42	5·37	5·33	5·88

For speeds below 14 knots, the turbine boats are inferior to the others, but, as is evidenced by the Table, they show marked economy at all speeds above 14 knots. The saving under service conditions is, however, not so marked, as is evidenced by Table XXXII., which gives the results of several months' working.

In Table XXXIII. are given the results obtained by the steamers when running simultaneously but in opposite directions. In these runs, the conditions approximate

very closely to the same for both boats when the average of a number of trips is taken. The results show very little difference between the turbine and reciprocating-driven

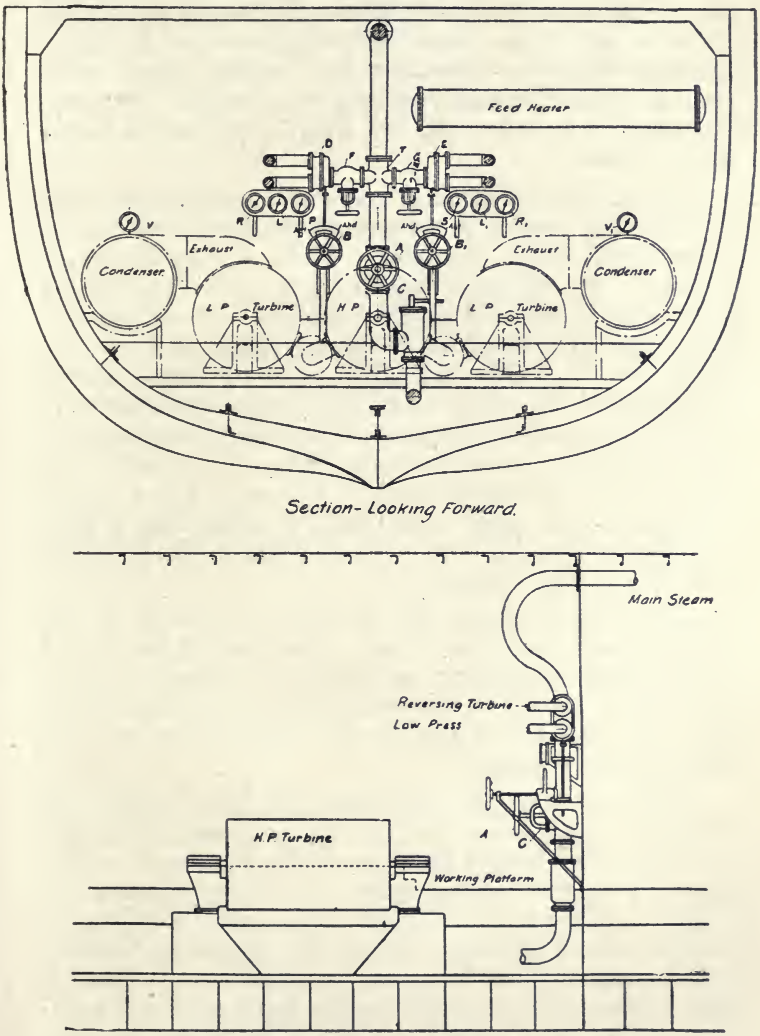


FIG. 178.—ARRANGEMENT OF TURBINES IN THE "TURBINIA."

boats, as regards either speed or coal consumption. The turbine boats save about 5 gals. of oil per trip, as compared with the reciprocators, and there is also a saving

of the engine-room staff. The turbine boats suffer, however, in manœuvring power. One interesting point brought out by Mr. Gray in his paper before the Institution of Naval Architects in July, 1905, was that although there was a considerable increase in the net tonnage of the turbine boats over the reciprocators—the gross tonnage being the same—yet this tonnage space being directly over the turbines was not of use for cargo carrying, and yet would in some cases increase the tonnage dues.

A consideration of the steam consumptions for marine reciprocating engines suggests that considerable improvements ought to be possible. Thus, for instance, the engines of the "Topaze"—which was tested against her sister turbine-driven cruiser "Amethyst"—took 20lbs. of steam per I.H.P. hour at 20 knots, 19lbs. at 18 knots, 18·8lbs. at 14 knots, and 23·7lbs. at 10 knots. Such consumptions would be considered very second rate in land practice, and it is impossible to imagine that marine-engine builders will always lag behind the builders of engines for stationary purposes. Of course better results than the above have often been obtained by marine reciprocators, but still there does seem to be room for considerable improvement.

Fig. 172 shows us the plan and elevation of the turbine-rooms for these boats. The general lay-out of the turbines and their auxiliaries is clearly shown. Fig. 173 gives us a cross-sectional view of the same. Figs. 174 and 175 illustrate the engine-rooms for the reciprocating engines.

It will be noticed that there are three shafts, the centre one being driven by the high-pressure turbine and the two outers by the two low-pressure turbines, which share between them the steam from the high-pressure turbine. These two wing shafts also carry the reversing turbines which normally run in the condenser vacuum so as to reduce their frictional resistance when running idle. These reversing turbines are made with comparatively few rows of blades, so that although they are less efficient than the main turbines, they cost much less, and their lack of economy is of no consequence. For manœuvring, the steam must be admitted in rapid succession to

the forward and astern turbines. A piston valve is usually used, but as it is difficult to maintain quite steam-tight, Mr. C. A. Parsons has used the valve illustrated in Fig. 176. The two belts D and F connect to the forward and astern turbines respectively, and both connect through ports to the valve chamber, and thence via G to the steam supply. The piston is an easy fit and is stepped. It has two hardened-steel faces N and K, which bear against seatings O and M, and prevent leakage of steam.

TABLE XXXIII.—*Showing Results obtained by Steamers Running Simultaneously, but in Opposite Directions.*

	Reciprocating Engine.	Turbine.
	"Antrim."	"Londonderry."
Number of trips	48	48
Average coal per trip (tons).	35·6	35·3
Average speed in knots ...	19·7	19·5
	"Donegal."	"Londonderry."
Number of trips	42	42
Average coal per trip (tons).	36·0	36·9
Average speed in knots ...	19·2	19·8
	"Antrim."	"Manxman."
Number of trips	29	29
Average coal per trip (tons).	38·6	38·6
Average speed in knots ...	19·05	20·3
	"Donegal."	"Manxman."
Number of trips	39	39
Average coal per trip (tons).	38·7	40·2
Average speed in knots ...	19·3	20·3

Fig. 177 shows the arrangement of Parsons turbines adopted in the steam yacht "Tarantula," which is interesting because the hull and power are similar to those of a standard torpedo boat. In this case, there are three separate expansion cylinders, the low-pressure being on the centre shaft.

Fig. 178 shows the arrangement of turbines on the "Turbinia," a passenger steamer on Lake Ontario,

Canada. A is the hand wheel controlling the main throttle valve C; B and B_1 are operating valves for admitting steam to the low-pressure and astern turbines for manœuvring purposes. Valves F and G are simple stop valves inserted in the branch pipes leading to the outer turbines, and are, of course, open during manœuvring operations. The gauges R V L F S and P are steam-pressure indicators. They also serve to indicate which turbines are rotating and how, there being no visible moving parts as in ordinary reciprocating practice. The

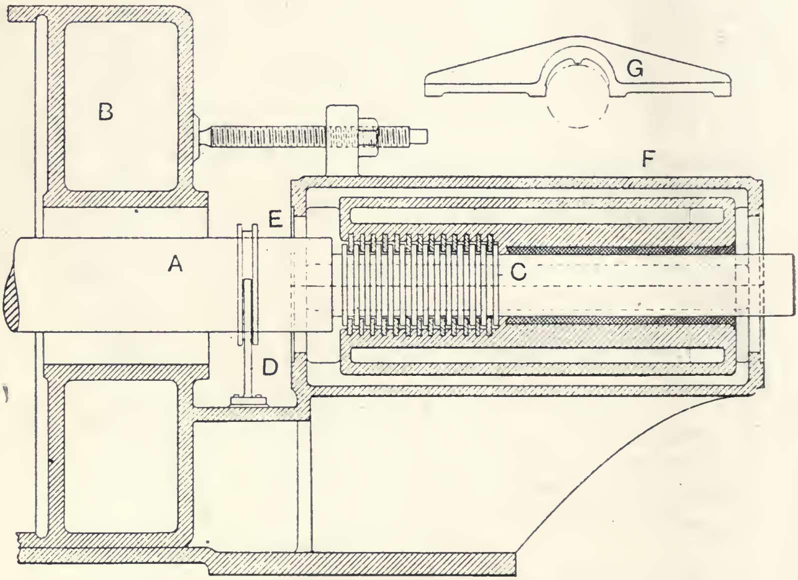


Fig. 179.—Thrust Block of "Tarantula" showing Method of Adjusting Blade Clearances.

high-pressure turbine on the centre shaft is usually cut off by the valve C during manœuvring.

The propeller thrust may be partially, and in most cases entirely, balanced by the end thrust in the turbine itself. As we have already pointed out in the chapter on blades, the blade end thrust is inversely proportional to the steam velocity.

By adopting a sufficiently low blade speed and suitable blade angles it might be possible to balance the propeller thrust by means of the end thrust on the blades. Such a

balance is not necessary, since by exposing a step on the drum of the rotor to the pressure of the steam, the total end thrust on the rotor can be made practically as large as desired, and certainly in all cases large enough to balance the propeller thrust. In any case, a small thrust block is required for adjustment purposes.

Fig. 179 shows the thrust block of the "Tarantula." The bearing is in halves, the lower half taking the ahead thrust, and the upper half the thrust when going astern. The thrust collars in the bearing are brass rings. Fig. 179 also shows the method of adjusting the relative positions of rotor and casing. The index *D* is bolted to the casing, and its distance from the collars E on the shaft accurately adjusted. The radial clearance over the balde tips is determined by the gauge G, which is placed across the top of the bottom half of the bearing, after the top portion has been removed.

Figs. 180, 181, and 182 illustrate the turbines of the "Carmania." In Fig. 182 is illustrated the low-pressure and astern rotor. The exhaust takes place at the change in diameter. The low-pressure section of the rotor is 11ft. 6in. in diameter by 2·5in. thick.

The chief obstacle to the universal use of turbines instead of reciprocating engines on board ship is the propeller, and hence it may be desirable to consider briefly in what manner the propeller affects the turbine design. The propeller, although frequently called a screw, is radically different from a screw working in a solid nut. The screw is really a statical machine in which the form, not only of the screw, but also of the nut in which it works, is rigid, and, moreover, the two maintain at all times the same geometrical relation. The marine propeller is rather different in its action; it is really a turbine driving the water away from ship. A certain amount of momentum is given to the water, and an equal amount of thrust impressed on the ship. Thus, if

V = absolute velocity of ship in feet per second;
v = absolute velocity of water in which the propeller works;
P = absolute velocity of advance of the propeller, that is the speed with which the ship would advance if the propeller were a screw rotating in a fixed nut, the water having no velocity.

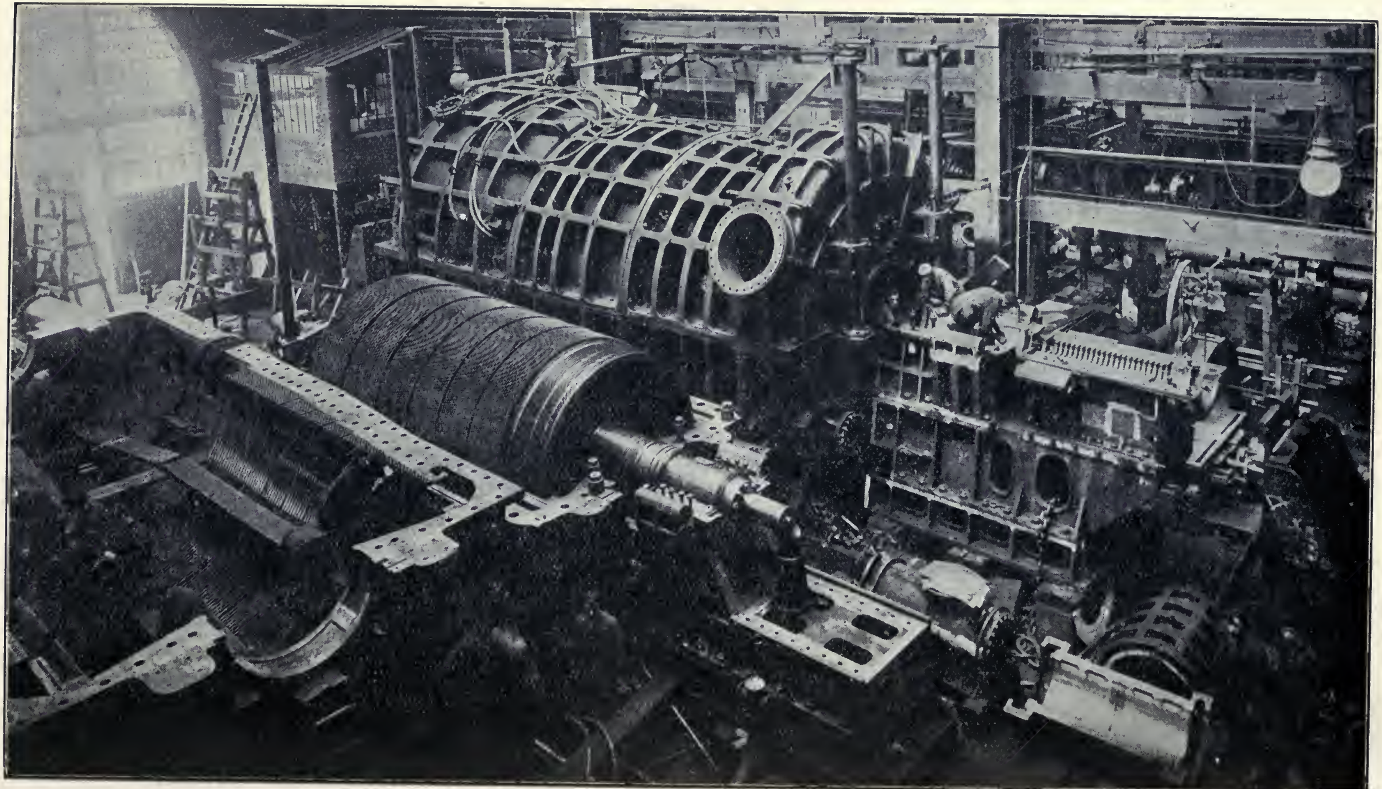


FIG. 180.—L.P. CYLINDERS AND H.P. ROTOR OF TURBINES OF "CARMANIA."

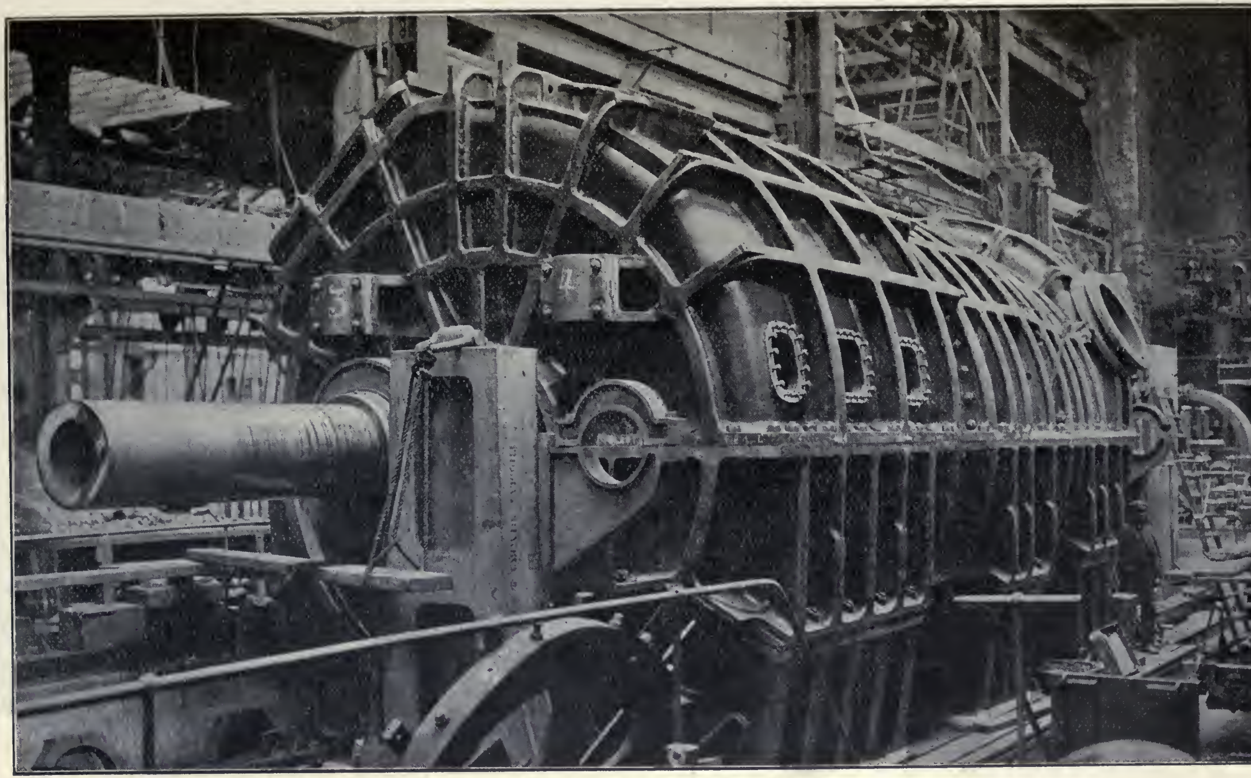


Fig 181.—Boring Out L.P. Cylinder of Turbine for "Carmania."

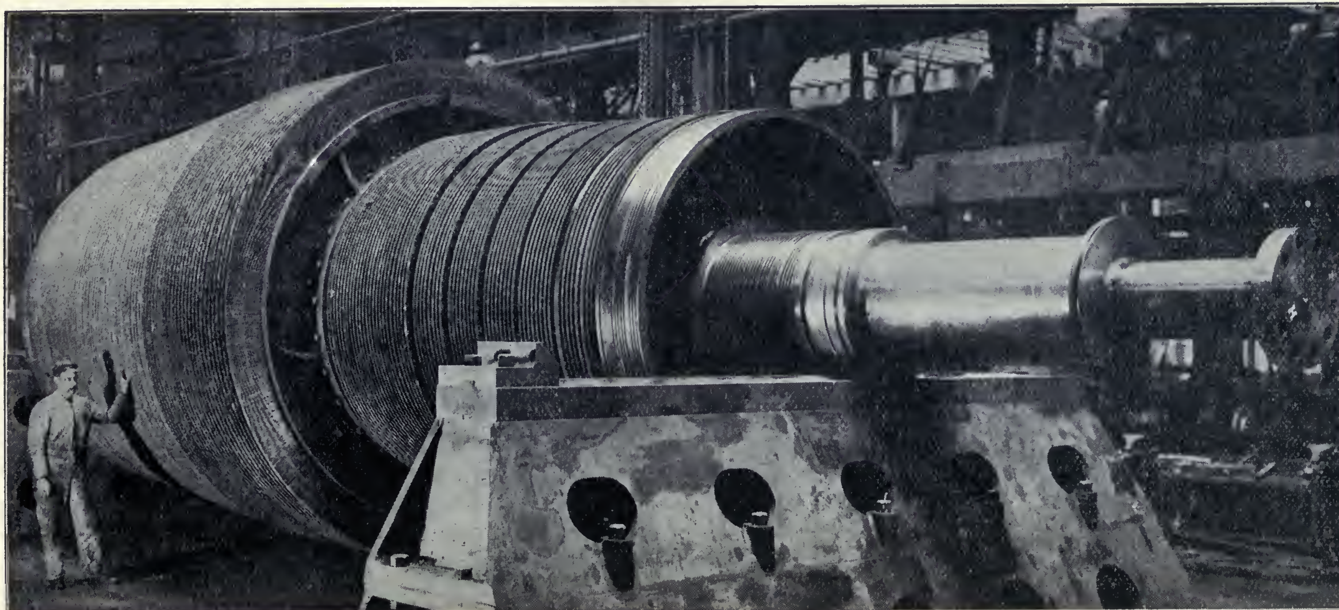


Fig. 182.—L.P. and Astern Rotor Being Balanced on Knife Edges (Turbines of "Carmania").

Then we know that V is usually less than P, and the quantity $P - V$ is the velocity of *apparent slip*, or, when expressed as a percentage of the ship's speed, the apparent slip. Now, if the water in which the propeller rotates were stationary, this would also be the true slip, and indeed it is usually spoken of merely as the slip. Generally, however, the water is not stationary, but is moving forward with the ship. The chief reason for this motion is the frictional drag of the ship on the water. At the mathematical surface of contact with the ship, the water clings to the ship's bottom and moves with it; but as we recede from the ship the velocity of the water becomes less and ultimately zero. There is another cause of this forward motion imparted to the water. As the ship advances its bow parts the water (Fig. 183) and

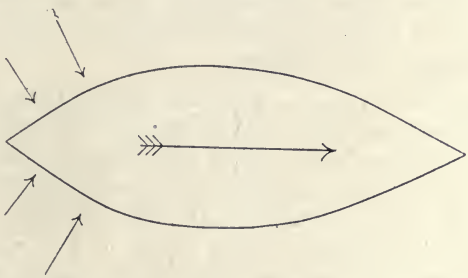


FIG. 183.—VELOCITY IMPARTED TO WATER AT STERN OF SHIP.

imparts some forward velocity to the water, but as the water closes round the stern the normal pressure of the water on the ship gives the water a velocity which, as indicated by the arrows, has a component in the direction of the ship's motion. The blunter the ship's stern the greater will this velocity be. At certain speeds the waves formed by the ship may be in such a position as to give to the water at the propeller a velocity with the ship (the motion of the water in a wave is oscillatory only, so that the wave position may be such as to retard the flowing wake of the ship). The true slip then is $P + v - V$, and is always positive, whereas if v is small enough the apparent slip may be negative, as indeed it often is. It may be pointed out now that the value of v, the water velocity, is different at different points both as to depth of water and position relative to the ship. This velocity

may be taken as the mean velocity of the water from which the propeller draws its supply.

The bulk of the velocity given to the water may be acquired before the water reaches the propeller. This is due to a suction action on the part of the propeller, on account of the blade shape not being such as to cut the water without shock when the water is moving with the unchanged velocity v. For instance, suppose that the water has a forward velocity with the ship of 5ft. per second, and that the propeller gives it a backward velocity of 3ft. per second just before it reaches the propeller blades. This change in velocity is due to a suction action on the part of the propeller.

This means that the front face of the propeller is a centre of reduced pressure and consequently draws water, and air if near the surface, from all round it. If air is drawn in at the surface the efficiency of the propeller will be considerably reduced.

In Fig. 184 we have a diagram showing what ought to occur at the propeller. A B represents the water velocity (relative to the ship's stern) at entrance, and is equal to $V - v$. C B is the circumferential velocity of the blade, and C E the axial—parallel to the ship's motion—velocity as it leaves the blade, and is equal to P. For a single screw placed over the keel line A B should be perpendicular to C B, but it is not necessarily so for a wing propeller. C E ought to be at right angles to D E, so that completing the parallelogram C B E D, the angle C D E should be the outlet angle of the blade, and we have

$$\tan CDE = \tan \beta = \frac{CE}{DE} = \frac{\text{pitch}}{\pi \times \text{diameter}}.$$

Clearly, since the pitch of the screw is constant at all diameters, the angle θ will vary as we move from the root of the blade to the tip, just as it actually does in the ordinary propeller (Fig. 185).

The thrust on the water per pound is

$$\frac{1}{g}(CD \sin \beta - AC \sin \alpha) = \frac{1}{g}(CE - AB) = \frac{P + v - V}{g}$$

when AB is at right angles to CB.

Now, if A is the area in square feet swept by the propeller less the boss, and if the velocity of the water crossing

this area is the same at all points in it, the weight of water acted upon by the propeller per second is

$$W = A \times \text{axial velocity} \times \text{weight per cubic foot.}$$
$$= 64\ A\ P \ldots\ldots\ldots \text{pounds.}$$

So that the propeller thrust is approximately in pounds,

$$t = \frac{64\,A\,P\,[\,P-(V-v)\,]}{g} = 2\,A\,P^2 - 2\,A\,P\,(V - v).$$

The area of the propeller blades as projected on a plane perpendicular to the shaft is some fraction of the disc area, or the area of the circle swept out by the propeller. According to Mr. Speakman* this fraction

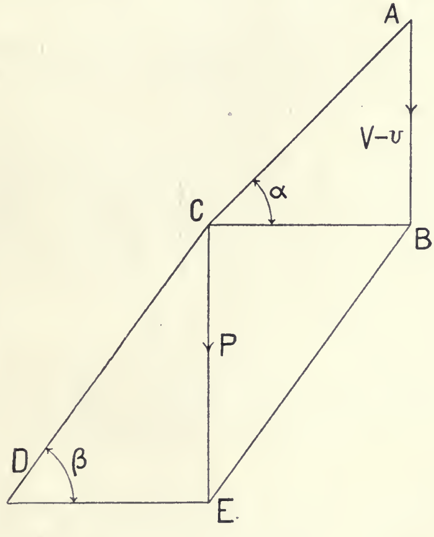


Fig. 184.—Velocity Diagram for Propeller.

depends on the class of boat and type of propelling machinery, the usual values for naval work being between 0·22 and 0·26; for ordinary reciprocating practice between 0·22 and 0·33; for torpedo-boat destroyers between 0·33 and 0·4; and above 0·4 for turbine work, although the value seldom exceeds 0·58.

* "Dimensions of Steam Turbines for Marine Work," by E. M. Speakman, Institution of Engineers and Shipbuilders of Scotland, October 24th, 1905.

If k is the ratio of the projected blade area to the disc area less the boss—not quite the same ratio as that mentioned just previously—then the thrust per square foot of projected blade area is, in lbs.,

$$t = \frac{2P^2 - 2P(V - v)}{k}.$$

This should not, apparently, exceed 1,500 or 1,800, or (pounds per square foot) although considerably higher values have been reached, the higher values being usually accompanied by high peripheral velocities, as is illustrated by Fig. 186, which is taken from Mr. Speakman's paper. The same writer gives the usual thrust in pounds per square foot as: For cargo boats, about 700lbs. or 850lbs.; for ocean mail boats, 850lbs. to 1,000lbs.; for cross-Channel boats, 1,100lbs. to 1,200lbs.; battleships and cruisers, 1,150lbs. to 1,500lbs.; and from 1,200lbs. to 1,600lbs. for torpedo craft.

A glance at the above expression for t shows us that the slip increases, for a given ship's speed, with the thrust per square foot, and therefore the kinetic energy imparted to the water in the propeller race is increased. This kinetic energy is clearly equal to (Fig. 184)

$$K = \frac{W}{2g}\left[P^2 - (V - v)^2\right] \quad . \quad . \quad . \quad . \text{ foot-pounds.}$$

The work done in propulsion is equal to the product of the thrust and speed, that is, equal to $T\ V$. The efficiency *of the propeller* is—neglecting friction and eddies—

$$E = \frac{T\ V}{K} = \frac{2\ A\ P\ V[P - (V - v)]}{A\ P[P^2 - (V - v)^2]} = \frac{2\ V}{P + V - v}.$$

This is equal to unity when the slip, which is equal to $P + v - V$, is zero, as we should expect. In the particular case when the water velocity v is zero the efficiency is

$$E = \frac{2\ V}{P + V}$$

Centrifugal Force Imparted to the Water.—Exaggerated notions are sometimes held as to the amount of centrifugal force imparted to the water by the propeller. It must be remembered that the propeller is not a centrifugal pump in which the water is travelling in the direction of rotation.

On the contrary, the water is travelling across the blades, and, if they are correctly shaped, at right angles to the plane of rotation. In practice, this water path will not be quite a straight line, but will be curved. This curvature means that the water has received a certain velocity in the direction of rotation, and hence is acted upon by a centrifugal force. As we have no means of estimating the amount of this curvature we cannot calculate the

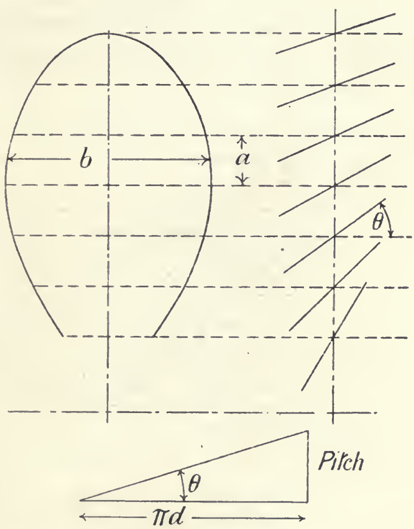


FIG. 185.—SHOWING BLADE ANGLES AT DIFFERENT DIAMETERS, THE PITCH BEING CONSTANT.

centrifugal force imparted to the water. When the propeller is not working under normal circumstances, as, for instance, when the ship is starting-up or reversing, the curvature of the water path across the propeller disc is bound to be great and the centrifugal force imparted to the water will be correspondingly great; and if the propeller be not too deeply immersed, will throw the water up under the ship's stern. This phenomenon can be witnessed any day at a landing stage.

Effect of Friction.—So far we have left out of account the friction of the propeller. It is usually, however, a very serious cause of loss, so that it deserves some consideration. The frictional resistance of the water on a small

portion of the propeller surface is proportional to the square of the velocity past the surface.

Consider a small portion of a propeller moving (Fig. 187) with a peripheral velocity H feet per second and making an angle θ to the plane of rotation. The friction of the water on this area is F along its surface, and has two components, one, $F \cos \theta$ in the plane of rotation resisting the rotation of the propeller and the other $F \sin \theta$ parallel to the propeller shaft, and resisting the motion of the ship. The works spent in overcoming these two resistances are, in foot-pounds per second, $H\,F \cos \theta$ and $V\,F \sin \theta$ respectively.

The velocity of the water past the surface is (Fig. 184) approximately the mean between $A\,C$ and $C\,D$, or, say, V

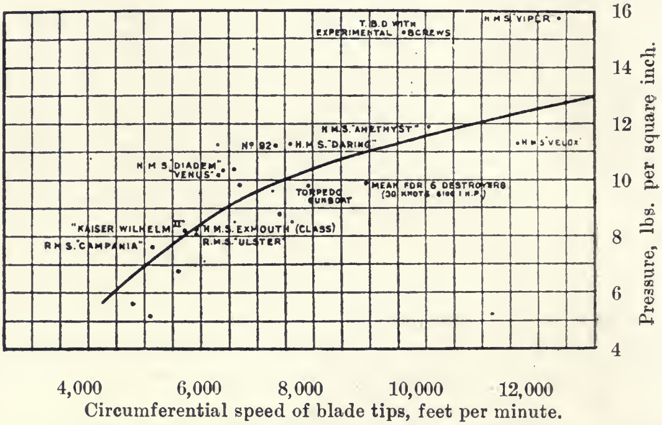


FIG. 186.—THRUST PRESSURES PER SQUARE INCH OF PROJECTED BLADE AREA, CORRECTED TO 12IN. IMMERSION.

divided by $\sin \theta$, where θ is the mean inclination of the surface. Then the value of F per square foot of propeller surface is equal to

$$c \frac{V^2}{\sin^2 \theta}$$

where c varies between $0 \cdot 004$ and $0 \cdot 008$ according to the state of the blade surfaces.

It is not difficult to estimate the value of this friction for the whole propeller. Thus, referring to Fig. 185, develop the surface of one blade (that is, lay it out flat),

and divide it into a number of parallel bands—between 5 and 10, according to the accuracy desired. Suppose one band is b feet wide and a feet high, then its area is $2\ a\ b$, since it has back and front faces. If r is the distance of this band from the axis of the shaft and p the pitch of the propeller, then the value of θ for this particular band is such that

$$\tan \theta = \frac{p}{2 \pi r}.$$

In order to illustrate this frictional loss, we have made a *rough calculation* of the friction losses for the screw shown in Fig. 188. The results are tabulated in Table XXXIV. The particular case represents a large ocean

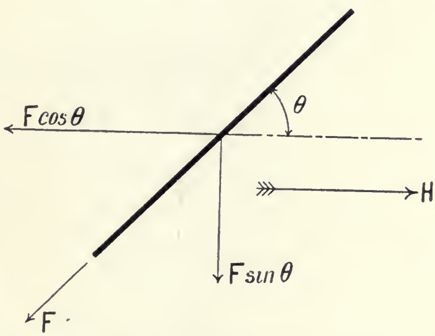


FIG. 187.—FRICTION ON PORTION OF PROPELLER BLADE, SHOWING COMPONENTS.

liner travelling at 20·7 knots and developing about 23,000 i.h.p., with a single 4-bladed propeller. The work spent in overcoming this propeller friction (apart from the friction of the boss) is roughly 12·7 per cent. of the effective propulsive horse-power, or 7·75 per cent. of the indicated power of the engines. The effective thrust per square foot of projected blade area is 880lbs., or 6·1lbs. per square inch. It is almost certain that in this case two propellers of smaller diameter would be more efficient than the single large propeller used.

Loss of Kinetic Energy.—From the formula for the thrust we calculate that the water velocity v is 0·42ft. per second, so that we have that the efficiency is

$$E = \frac{2 \times 35}{40 + 35 - 0{\cdot}42} = 93{\cdot}9 \text{ per cent.}$$

TABLE XXXIV.

Ship's speed, V = 35ft. per second; revolutions per minute = 75; diameter of propeller, 26ft.; pitch, 32ft.; four blades; effective propulsive horse-power = 14,000; coefficient of friction, $c = 0 \cdot 005$.

Mean Radius in Feet r.	Breadth b.	Height a.	Area = Approx. $2\,ab$.	$\tan\theta = \frac{p}{2\,\pi\,r}$.	θ, degrees.	Sin θ.	Sin $^2\theta$.	Cos θ.	Friction = $F = \frac{c\,V^2 2\,a\,b}{\sin^2\theta}$.	$F\,H \cos\theta$.	$F\,V \sin\theta$.	H, in Feet per Second.
2·9	7·4	2·9	43	1·76	60	0·87	0·76	0·5	350	4,000	10,600	23
5·8	8·7	2·9	51	0·88	41	0·66	0·44	0·75	710	24,400	16,400	45
8·7	7·8	2·9	45	0·59	30	0·5	0·25	0·87	1,100	65,000	19,300	68
11·5	5·0	2·9	29	0·44	24	0·4	0·16	0·91	1,100	90,000	15,400	80
Totals	per	blade ...	168	...	...	...	...	...	...	183,400	61,700	...

Friction work per blade per second ... 245,100 foot-pounds.

Work spent in friction on propeller blades = 980,000, or $\frac{980{,}000 \times 100}{14{,}000 \times 550} = 12 \cdot 7$ per cent. of effective power.

Hence, 6·1 per cent. of the effective propulsive horse-power, or equal to 850 horse-power, is lost in the wake. This is equal to 3·7 per cent. of the indicated horse-power of the engines. If we assume that the mechanical efficiency of the engines and shafting is 81 per cent., we have the following distribution of power :—

	H.P.	Per cent.
Effective propulsive h.p.	14,000	60·85
Friction of propeller	1,780	7·75
Loss in wake	850	3·7
Loss in shafting and engines	4,370	19·00
Eddy losses at propeller........	2,000	8·7
Total	23,000	100·00

The above propellor is not very well proportioned. It has too much surface. Three blades would be sufficient,

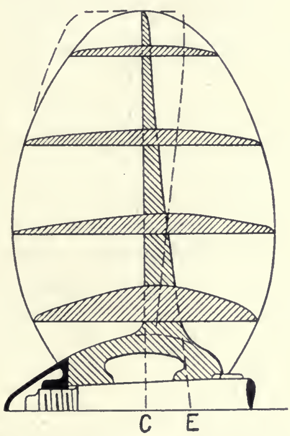


Fig. 188.—Propeller, Developed Blade.

or else four narrower ones. A somewhat smaller diameter and a higher rotational speed would probably give better results.

Eddy Loss.—This is due mainly to incorrect blade shape and to the thickness of the blade, which produces some eddies when parting, and, more particularly, when closing together the water. This latter loss can be reduced by using thin propeller blades, which must then be made

of steel, phosphor-bronze, gun-metal, or some other such material.

Thus referring to Fig. 188, the thickness of the blade projected to the shaft is often

$$CE = K\sqrt{\frac{d^3}{b\,n}}$$

where

d = diameter of shaft in inches.
b = maximum breadth of blade, inches.
n = number of blades.

K is a constant equal to about $6 \cdot 3$ for cast iron, $3 \cdot 2$ for gun-metal, $2 \cdot 5$ for cast steel, and $2 \cdot 35$ for the best bronzes.

In order that the blade may have the correct shape, the blade angle at inlet should be less than at outlet. Since the blade angle is the mean angle between the back and front faces meeting at the edge, it follows that this is secured to some extent with the ordinary flat front-face construction. Thus referring to Fig. 184 we see that we should have

$$\frac{\tan \alpha}{\tan \beta} = \frac{V_r - v}{P}$$

Wing Propellers.—In the case of wing propellers, the water seldom enters parallel to the shaft, because of the curvature of the ship's run. Consequently, there is bound to be considerable eddy loss on this account, seeing that this curvature would require a larger difference between the inlet and outlet angles when the blade is above the shaft, and a less difference when below the shaft, or vice versâ.

If the propeller blade has too large an angle there will be negative pressure on the backs of the blades at the entering side due to eddies formed as is illustrated in Fig. 189. This is commonly known as cavitation. Owing to the reduced pressure thus produced a suction effect takes place which increases the water velocity and thus partially reduces the cavitation.

Propeller Proportions.—The usual shape of the propeller blade is approximately elliptical, although we have no certain knowledge that this is the best form. In order to increase the ratio of projected to blade

area for turbine propellers, the breadth of the blade is sometimes much increased towards the tips. In itself this is good, as the tips of ordinary blades are too far apart to effectively control the flow of all the water in the propeller circle. Unfortunately such blades are weak unless made very thick at the roots, and this reduces the efficiency. Just what is the best form it is difficult to say. We may yet see propellers fitted with several narrow blades

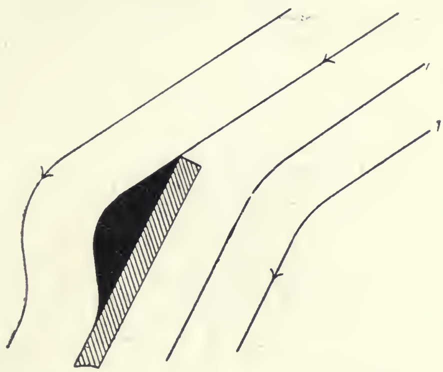


FIG. 189.—EDDY LOSS (SHOWN BLACK) DUE TO INCORRECT BLADE ANGLE.

externally supported by a circumferential ring. Turbine propellers should be experimentally balanced, and their surfaces given a clean, smooth finish.

Froude gives the best area of the blades in square feet as being

$$A = 8{\cdot}9\frac{R}{v^2}$$

where

R = resistance of ship in pounds.

v = speed of ship in feet per second.

He calculated mathematically that the best pitch ratio (ratio of pitch to diameter of blade circle) is about 2. For turbine work this has to be much reduced, as otherwise the revolutions would be too low. For most of the larger turbine steamers so far built the ratio is about 0·9 and somewhat higher in fast warships.

The peripheral speed of the blade tips is usually between 10,000 and 11,000ft. per minute on turbine boats, but in the "Carmania" and "Victorian" it falls

to about 8,000ft. per minute, and even less. The corresponding speed for the ordinary Atlantic liner is about 5,000 or 5,500ft. per minute.

According to Speakman, the diameter of a propeller in feet is equal to

$$\frac{\sqrt{\text{effective thrust in pounds}}}{C}$$

The value of the coefficient C varies, apparently, from about 22 for large ocean boats to about 27 or 29 for cross-Channel boats, and as high as 30 or 31 for fast naval ships.

Effect of Bad Weather.—It is found in practice that turbine-driven boats do not show the same superiority over those having reciprocating engines when at sea that might have been expected from the trial trip results. The reason for this is pretty clear.

Owing to their high rotational speed turbine-driven propellers are of smaller diameter than those for use with reciprocating engines. The propeller thrust is not appreciably influenced by the type of engine adopted, and is equal to the product of the slip and the mass of water acted upon by the propeller per second, which latter quantity is proportional to the product of the area of the propeller circle and the speed of propeller advance. Clearly, then, if the area be reduced we must increase the speed of advance or, in other words, we must increase the slip.

In so far as the ordinary propeller blades are not wide enough to effectively control the flow of all the water passing through the blade circle, an increase in the width may somewhat reduce the slip by increasing the effective area of the propeller race. Still the fact remains that the slip must be much higher when turbines are used than in other cases.

The effect of this on the speed of the ship in bad weather is easily seen from the equations previously given.

In order to fix our ideas, the example represented in Fig. 190 has been worked out. The full lines refer to a ship having a slip of 11·6 per cent. at full speed, and driven—presumably—by reciprocating engines. The

dotted lines refer to a turbine boat of the same size and speed, but having a slip, at full speed, of 35·5 per cent. This latter figure is unduly high, but will serve to illustrate our point.

Both ships develop the same effective propulsive horse power, which is, moreover, constant in all weathers. The normal speed is 20 knots, but this is reduced by bad weather. Fig. 190 illustrates how the speed of propeller advance P and the propeller thrust T are affected by a decrease in the speed of the ship on account of head winds or rough seas.

The figure illustrates how the turbine boat suffers in bad weather. Thus, if the head winds increase the total resistance of the ship to 160 per cent. of its normal value, we see that the speed of the turbine boat is reduced to 5 knots, and the other to 10·5 knots. Since, however, the thrust is to a marked extent dependent on the speed, the above figures are unduly favourable to the reciprocating engine. In a given seaway and wind the resistance at 5 knots would be less than at 10·5 knots.

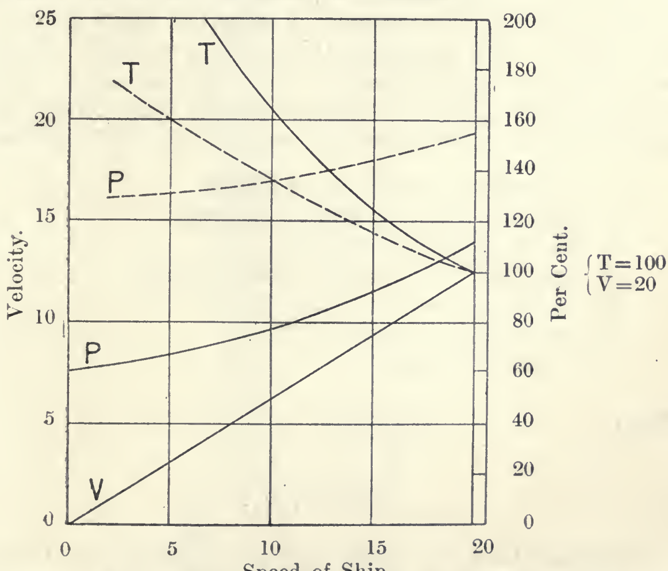


Fig. 190.—Effect of Weather on Speed of Ship.

V = Speed of ship.
P = Speed of propeller advance.
T = Thrust of propeller.

Further, the deeper immersion of the smaller propellers is somewhat in their favour. The effect of friction and eddies might also influence the results, but the fact remains that the turbine boat must suffer a greater loss of speed in rough weather than a boat driven by reciprocating engines.

For ocean-going turbine boats it would seem that a small pitch ratio—even though accompanied by higher frictional resistance, and not giving such a good showing on trial—is advisable in order to keep down the slip.

Balance of Propeller Thrust.—It is often claimed that the turbine thrust will balance the propeller thrust. In general, however, only a partial balance is obtained. The lower the axial velocity of the steam the greater the end thrust of the turbine. The limitations imposed on the turbine end thrust are best considered by means of an example. We will suppose that the frictional drag on the blades is just balanced by the reaction thrust on the blades due to the progressive increase in the axial velocity of the steam in the sections of constant blade height—produced by the increase in the steam volume without any corresponding increase in the passage area. Considering one pound of steam passing per second and neglecting the pressure on the steps of the drum we have

A = annular area swept by blades.
V = speed of ship in feet per second.
v = axial velocity of steam.
k = efficiency of propeller and shafting.
W = indicated work of turbine.
$k\,W$ = work done by propeller.
P = propeller thrust.

Then
$$P = \frac{k\,W}{V}$$
$$= \frac{0{\cdot}6\,W}{V}, \text{ say.}$$

Suppose that the turbine is working between 170lbs. and 1·5lbs. absolute, the steam volumes at those pressures being 2·63 and 190 cub. ft. Suppose also that the steam velocity is constant—as for instance, when using one

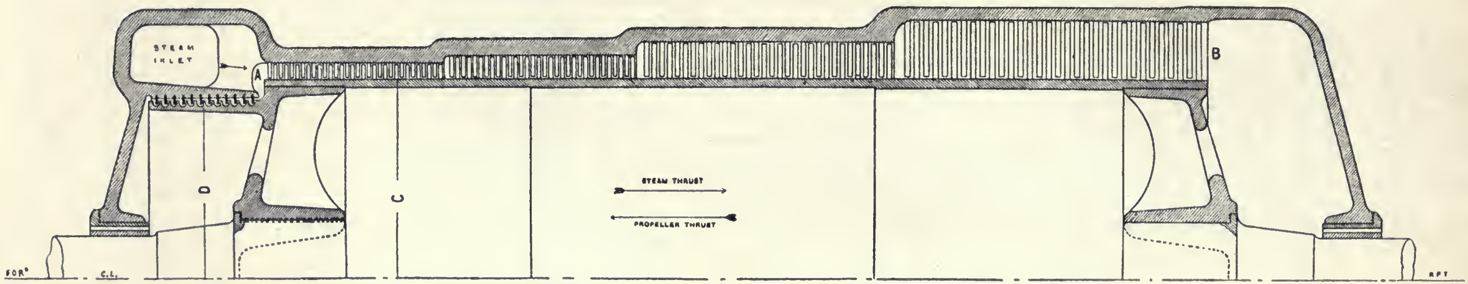


FIG. 191.—SECTIONAL ELEVATION OF H.P. CYLINDER OF A PARSONS TYPE OF MARINE TURBINE.

high-pressure and two low-pressure turbines of the same diameter—then the turbine thrust is

$$T = \frac{K}{4}\frac{(p_1 A_1 - p_2 A_2)}{(1 - m)}$$
$$= \frac{K (p_1 u_1 - p_2 u_2)}{v \, 4 \, (1 - m)}$$

Assuming that $\frac{K}{4 \, (1 - m)}$ is equal to unity, and inserting values—pressures in pounds per square foot—we see that

$$T = \frac{144 \, (447 - 285)}{v}$$
$$= \frac{23{,}300}{v} \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{lbs.}$$

If now $W = 140{,}000$ foot-pounds (corresponding to a turbine efficiency of 60 per cent.), we see that

$$P = \frac{84{,}000}{V} \quad . \quad . \quad . \quad . \quad . \quad \text{lbs.}$$

For a complete balance we must have

$$\frac{v}{V} = \frac{23{,}300}{84{,}000} = 0 \cdot 36.$$

For instance, if the ship's speed is 21 knots, V is 35·5, and for a balance the steam velocity would only be 12·8ft. per second, an absurdly low value.

An approximate balance can usually be obtained between the propeller and turbine thrust by allowing the steam pressure to cause a thrust on a step, or increase in diameter, on the rotor. Such an annular step is shown at A, the high-pressure inlet end of the rotor in Fig. 191.

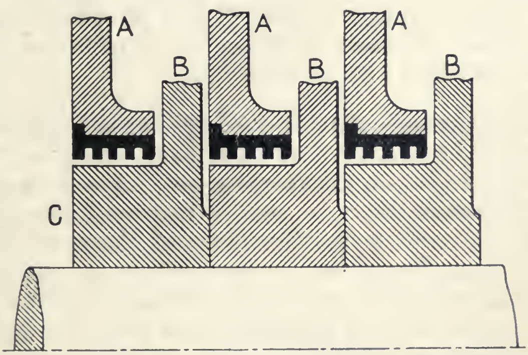
This balance of the propeller thrust is a matter of considerable importance, as it somewhat increases the efficiency of the shafting, reduces the quantity of lubricating oil required, and reduces the chances of trouble at the thrust block.

Since in an impulse turbine there is no end thrust on the blades, it becomes necessary to use the ordinary thrust block, or some kind of revolving balance piston. Fig. 192 shows a method suggested by the author. A and B are respectively diaphragms and wheels, the latter

being carried by the shaft S. The chamber in which the wheel B is rotating is at uniform pressure, but the pressure on the left-hand side of the diaphragm A is greater than on the right-hand side. Consequently with the diaphragm encircling the boss of the wheel as shown there is an end thrust on the wheel B equal to the product of the area of the annular surface C and the difference between the pressures on the two sides of the diaphragm. This end thrust can be divided among the individual wheels or confined to one or two at the high-pressure end. The former method is the better.

The chief objection to this method is that it may in many cases require a large diameter over the hub of the wheel, and thus lead to considerable leakage loss past the diaphragm.

It may be pointed out now that it is the easiest thing in the world to design an impulse turbine with considerable end thrust, due to the static steam pressures on the rotor. Considerable care in the design is necessary if this end thrust is to be avoided—as in turbines for stationary



Centre Line of Shaft.

FIG. 192.—METHOD OF PRODUCING END THRUST IN AN IMPULSE TURBINE.

work. The static steam pressures on all shoulders on the shaft or wheels must be calculated and balanced against each other. In order to do this it is often necessary to enlarge the diameter of the shaft where it leaves the casing at the high-pressure end.

The thrust block on the propeller shaft of an ordinary reciprocating engine might be replaced by a balance piston of the type so common in turbine work. Such an arrangement would cause some little loss of steam, but might very well make up for it by the reduction in friction, and consequent increased mechanical efficiency, and the reduction in the quantity of lubricating oil required.

Variable Speed Turbines.—The power required to propel a ship varies approximately as the cube of the speed. The number of sets of moving blades (in series) in a turbine decreases with an increase in speed, being, in fact, inversely proportional to the square of the speed: Hence we want, if possible, some turbine arrangement which shall comply with these conditions, if the ship is intended to do much cruising at reduced speeds. The usual method of providing for a reduction in speed is by throttling the steam supply, but this is wasteful, and makes the steam turbine relatively less economical at low speeds than a reciprocating engine, as was evidenced by the steam-consumption figures of the Midland Railway Company's steamers. At 14 knots the steam consumption of the reciprocating-engine steamers is 46·5 per cent. of the consumption at 20 knots, whereas it is 50·6 and 54·2 per cent. for the two turbine steamers.

One method of securing good results at all speeds is to have several turbines of various sizes which can be connected—as to steam distribution—in various ways. The lower the speed the more sets of moving blades in series we require ; and hence the more of these cylinders (or turbines) must be placed in series. There are a great many difficulties, one of which is due to the fact that the relative areas of the turbine passages, if correct for the series arrangement, will not be correct when they are connected in parallel.

Another difficulty is that of adjusting the relative proportions of the turbines so that they shall give equal efficiencies at all speeds and also a symmetrical distribution of power to the propellers. Each case must be considered separately. We may illustrate by a simple example. Suppose that the full speed of a warship is

n times its normal cruising speed, and that the indicated horse-power required varies as the cube of the speed. Let there be three shafts, the centre carrying a small turbine; the outers carrying two exactly equal larger turbines which always run in parallel. At cruising speed the steam passes from the centre to the two outers. At full speed all three take steam at full pressure and exhaust into the condenser. When run in series, let the volumes of the steam in cubic feet per pound—including moisture—at entrance to the centre and outers be 2·26 and x, the former corresponding to 200lbs. per square inch absolute. When run in parallel the *ratio of the steam volumes* admitted will be unaltered, but all the steam will now be at full pressure and the volumes will be increased in the ratio of the speeds. Assuming that the horse-power is proportional to the steam consumption, we see that

$$\frac{n\,(2\cdot 26 + x)}{2\cdot 26} = n^3.$$

From this we can determine x. For instance, suppose the cruising speed to be half the full speed. Then n is two, and we find x to be 6·78, corresponding to a pressure of about 57lbs. per square inch absolute at the inlet to the outer turbines at half speed. The powers developed by *each* outer and by the centre will be about 35 and 30 per cent. of the total at half speed, and 37·5 and 25 per cent. respectively at full speed; the power developed by the two outers being, of course, the same. The above solution seems very satisfactory at first sight, but a little consideration shows that there are some exceedingly grave defects. The total expansion of the steam between stop valve and condenser should be about 100-fold, whereas the centre turbine only has provision for a *three*-fold expansion and the outers for a 30-fold expansion. Clearly then, although quite economical at cruising speed they will be anything but that at full speed, and indeed would not develop the required power. The above example will therefore serve to bring out one of the fundamental difficulties inherent in such a combination of turbines.

Fig. 193 shows the arrangement of turbines used by the Parsons companies where special provision has to be

made for cruising at reduced speeds. There are five separate turbines, A, B, C, D, and E. A is the high-pressure turbine, B and C the low-pressure turbines.

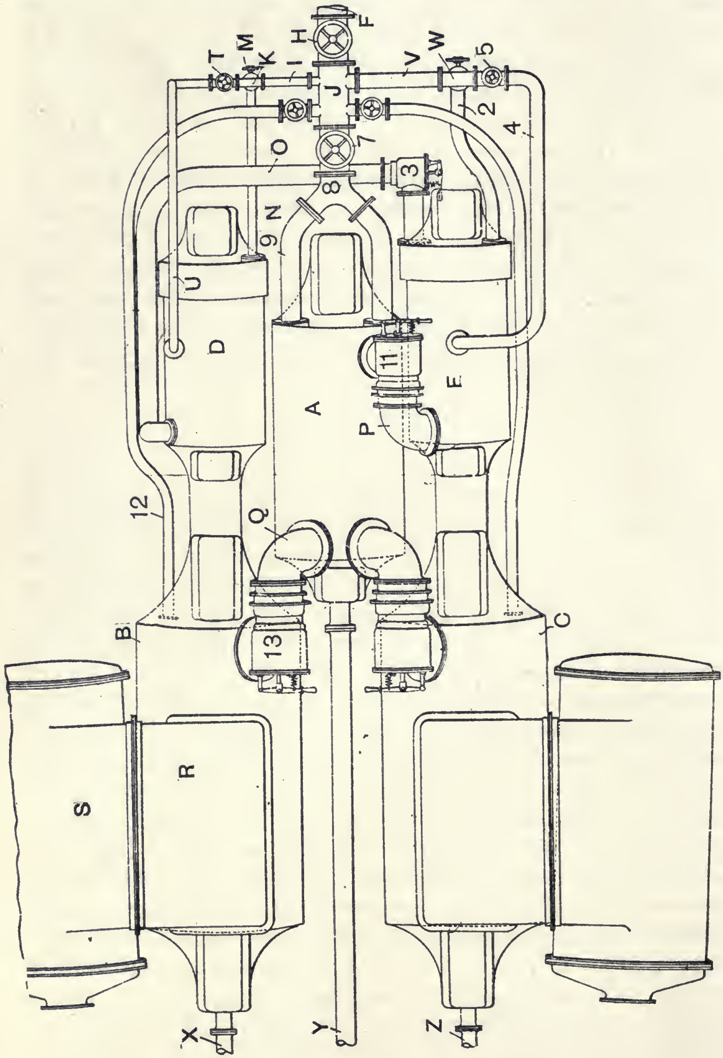


Fig. 193.—Parsons Marine Steam Turbine for Cruising at Variable Speeds.

D and E are cruising turbines. At cruising speeds, the steam passes through D, E, and A in order, then splits and passes through B and C in parallel, and thence to the

condensers S. The steam path for the above arrangement is F I K N to turbine D ; then by pipe O to turbine E ; then by way of P to the central high-pressure turbine A, and from there by way of the pipes Q into the low-pressure turbines B and C. It will be seen that by this arrangement a great many sets of moving blades are obtained in series as desired. It is, however, not possible to insure that the passage cross-sections in the different turbines shall be correct for both cruising and maximum speeds. If a somewhat higher speed is required steam is admitted by way of F I T U to a point part-way along the turbine D, the rest of the expansion being as before. For still higher (but not maximum) speeds, the steam is admitted by way of F V W 2 or F V M 5 4 to the turbine E, the non-return valve 3 preventing any back flow into the turbine D. For full speed steam is admitted by way of F J 7 8 9 to the high-pressure turbine A, and thence to the turbines B and C. The cruising turbines can be disconnected from the turbines B and C at full speed.

It is clear that this arrangement is very expensive and complicated, and in the majority of cases its commercial value is doubtful.

Probably more satisfactory results would be obtained by a combination of reciprocating engines and low-pressure turbines ; the latter only to be used at full speed. Such an arrangement, besides being economical at all speeds, would have the advantage of insuring good manœuvring qualities and doing away with the necessity of astern turbines. For instance, suppose we have reciprocating engines on the outer shafts exhausting (at full speed) into the turbine on the central shaft ; their respective powers being—to some convenient unit—100 each, or a total of 300. If now the cruising speed is 0·65 of the maximum, and the cruising power 0·35 of the maximum, each reciprocating engine will then develop 52·5, and since the speed is only 0·65 of the maximum the mean effective steam pressure will only be reduced by 19·2 per cent.

Reversing Turbines.—The usual method adopted to secure reversal in the direction of the propeller rotation is to employ separate reversing turbines fixed on the same shaft or shafts as the low-pressure cylinders, and revolving in the exhaust vacuum when the boat is going

ahead. This cuts down their frictional resistance, and makes a compact turbine system. These reversing turbines are comparatively simple, seeing that steam economy is not of the greatest importance, and first cost and size are matters of very great importance. The proportion of the full-ahead power obtainable from the reverse turbines depends on the requirements of the particular case; cross-channel and river steamers requiring a greater proportion than ocean boats. The Channel turbine steamer "Queen" was, for instance, stopped in 2·5 times her own length from steaming at 19 knots. The astern power is usually from one-eighth to one-third the ahead power, giving an astern speed of from about one-third to two-thirds of the ahead speed.

Fig. 194 illustrates the ordinary arrangement of combined low-pressure forward and astern turbines on one shaft. The two turbines both exhaust at H G into the condenser, the direction of flow of the steam in the forward turbine being from N to G, and in the astern turbine from M to H. When the forward turbine is running the rotor of the astern turbine is running in a vacuum. The forward end (N) of the rotor is held in position by a small thrust block. Owing to the considerable relative expansion between rotor and casing in large turbines of this type, the packing at the astern-turbine end has to be of a somewhat different character from the ordinary labyrinth packing. Its details will be discussed in a subsequent chapter.

The illustration of the low-pressure rotors of the "Carmania" will illustrate clearly the general arrangement. (See Fig. 182).

By suitably choosing the blade shapes the turbine can be made reversible. The passage areas must also be adjusted to suit the new condition of affairs. This can be secured by diaphragms which can be made to cover more or less of the guide openings as desired. Such a turbine will be uneconomical for both forward and backward running unless a very expensive construction is followed, which might better be replaced by cheap simple astern turbines.

Turbo Air Compressors.—The turbo air compressor is in principle a reversed turbine in which, however, the

moving blades must not be symmetrical. It is not necessarily a (reversed) turbine of the Parsons type. The effect of the moving blades is to impart kinetic energy to the air. This kinetic energy is converted into pressure energy by a suitable adjustment of the passage cross-sections. The whole of this compression might take place in the stationary blades, or part in the stationary and part in the moving blades ; which is inherently a better plan, as by this process of compression the average velocity of the air is less than in the former case for the same amount of compression. Or we might have the whole of the compression performed in the moving blades, the fixed blades acting merely as guides.

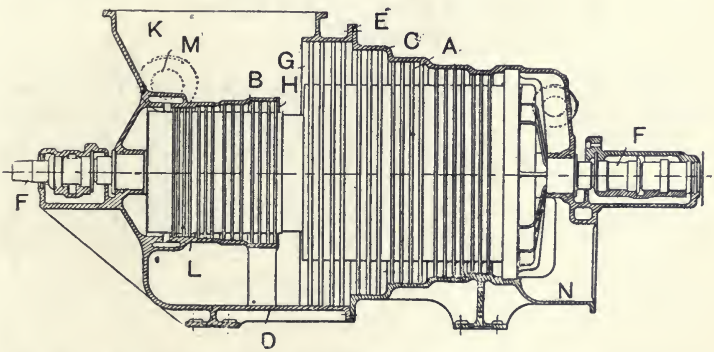


FIG. 194.—GENERAL ARRANGEMENT OF LOW-PRESSURE AND ASTERN TURBINE FOR MARINE WORK.

The action of a turbo-compressor may be indicated by a consideration of Fig. 195. Neglecting friction, we see that a particle of air at P is acted upon by a normal pressure from the blade, and hence will describe *a path in space* such as P R. The motion relative to the blade will then be obtained by compounding its velocity along P R with the velocity of the blade. The resultant motion is parallel to the surface of the blade, so that the moving blade will deliver the air to the fixed blades, which in their turn will pass it on to the next set of moving blades ; and so on.

It is not proposed here to go very fully into the details of turbo-compressor theory. Much of it is precisely similar to that explained in connection with steam turbines. The

method of determining the velocities, pressures, and volumes of the air at different points along the turbine may be briefly indicated.

In Fig. 195 we have the general velocity diagram.

$A\ B$ = inlet velocity relative to the moving blades.
$C\ E$ = outlet velocity relative to the moving blades.
$A\ C$ = absolute velocity at inlet to moving blades and outlet of stationary blades.
$B\ E$ = absolute velocity at outlet of the moving blades and inlet to stationary blades.
ϕ = inlet angle of stationary blades.
β = outlet angle of stationary blades.
α = inlet angle of moving blades.
θ = outlet angle of moving blades.
V = velocity of blades.

The (absolute) velocity B E at entrance to the fixed blades should, of course, not be less than the outlet velocity A C, as the converse would mean that there was an expansion in the fixed blades. Similarly, the inlet relative velocity A B must never be less than the outlet velocity or there will be an expansion in the moving blades.

The indicated work of compression is

$$\frac{V}{g}(A\ B \cos \alpha + C\ E \cos \theta).$$

In the above case *cos* θ is negative.

We may distinguish between three modes of compression.

Case I.—All the compression takes place in the moving blades. Then, since there is no compression and therefore no change of velocity outside the fixed blades we see that the absolute velocity of the gas remains unaltered throughout; that is

$$A\ C = B\ E$$

The work done in compression is

$$\frac{A\ B^2 - C\ E^2}{2\ g} \quad . \quad . \quad \text{foot-pounds.}$$

For a compressor of this class, the radial clearances for the moving blades must be small. Those for the fixed blades may be quite large, but if they are the fixed

blades should have a shroud and the axial clearances should be small to reduce the eddy losses between the flowing and the stagnant air. The friction losses would be somewhat higher than for a compressor of the third class, so that there seems to be no reason why this type should be preferred.

Case II.—No compression in the moving blades. All the compression takes place in the fixed blades, the moving blades merely generating kinetic energy in the gas. Consequently the relative velocity in the moving blades remains unaltered, so that

$$A\,B = C\,E.$$

The work done is

$$\frac{B\,E^2 - A\,C^2}{2\,g}.$$

In a compressor of this class the whole of the increase in pressure takes place in the fixed blades. Consequently the axial clearances should be small and also the radial clearances between the fixed blades and the rotor. In general structure, a compressor of this type corresponds to an impulse turbine, although the moving blades will not be symmetrical, save in special cases.

Case III.—Compression in both fixed and moving blades. This type of compressor is analogous to the Parsons turbine. The whole of the kinetic energy is generated in the moving blades, and converted into pressure energy in both fixed and moving blades. A C is less than B E, since there is compression in the fixed blades and A B is greater than C E since there is compression in the moving blades.

The compression in the fixed blades is

$$\frac{B\,E^2 - A\,C^2}{2\,g}.$$

and the compression in the moving blades is

$$\frac{A\,B^2 - C\,E^2}{2\,g}.$$

If these are equal we must have

$$A\,B = B\,E,$$

and

$$A\,C = C\,E.$$

The fixed and moving blades have the same angles, and the compressor is practically the same as a reverse reaction turbine. Its most efficient blade angles will be practically the same as those for the turbine.

Owing to the rise in pressure in the moving blades of compressors belonging to Class I. and Class III., these will have an end thrust tending to blow the rotor towards the low-pressure inlet.

The calculation of the number of stages is similar in nature to the case of a turbine. There is this difference, however, namely, that the fact that the machine has losses, causes an increase in the number of stages above the number required for a perfect compressor, whereas in a turbine the reverse is true. For instance, if the theoretical work of compression is 60 B.Th.U., and the hydraulic efficiency is 50 per cent., the actual work spent in the compression will be 120 B.Th.U. If then the indicated work in each stage is 5 B.Th.U., there will be 24 stages. For an air *turbine* with the same theoretical work and indicated work per stage, but with an hydraulic efficiency of 58 per cent., the number of stages would be only seven.

This great increase in the number of stages required for a compressor as compared with a turbine makes the compressor rather expensive.

So far as the author is aware all the turbo-compressors so far constructed belong to the third class. This is indeed probably the most desirable type for general work. The particular field of the turbo-compressor is that of supplying air at low pressure—a few inches of water—and approximately constant volume.

There are certain defects inherent in this form of compressor. The internal losses involve the production of heat, which by expanding the air increases the work of compression between two given pressure limits. These losses are greater than the comparatively small losses which occur in a reciprocating compressor. Then again, the leakage through the clearance spaces over the ends of the blades is bound to be very serious. In a turbine this leakage results in a *partial* loss of energy, part reappearing as available heat energy lower down the turbine. Just the same action takes place in the compressor, but here the partial reappearance (as heat

energy) of the pressure energy loss from higher up the turbine is not a source of gain, but on the contrary is a source of loss ; for it heats up the air, expanding it and increasing the amount of work required for its compression. Obviously this loss will become more and more serious the higher the pressure to which the air is compressed, thus imposing a limit on the pressures for which such compressors would be suitable. It is usual to water-jacket the cylinders of a reciprocating compressor, so as to make the compression as little like adiabatic and as much like isothermal compression as is possible. The turbo-compressor could, of course, be water-jacketed on both the casing and rotor. Still, a turbo-compressor

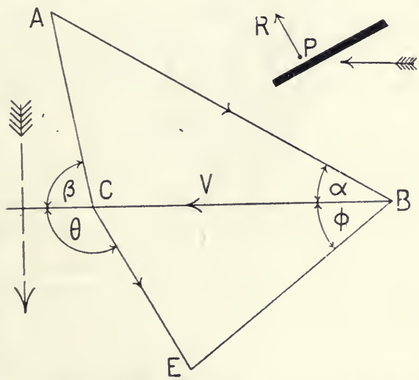


Fig. 195.—Turbo-Compressor Velocity and Blade Diagram.

cannot hope to rival the reciprocating compressor for high pressures. The turbo-compressor has some advantages however ; its mechanical simplicity, the absence of valves and the steadiness of the air pressure are strong points in its favour. The question of the efficiency of the compressor at light loads is an important one. Usually a constant pressure is required, necessitating a constant speed. The air may then be throttled at delivery, or a species of blast regulation similar to that used in the Parsons steam turbine may be adopted. In this latter case the delivery of air will be intermittent, thus destroying one of the advantages of the turbo-compressor.

A few economy tests have been made on turbo-compressors, but they are not reliable. According to

some tests at the Farnley Iron Works, near Leeds, a combined efficiency of 61 per cent. for a steam turbo-compressor or blowing engine delivering air at 3lbs. pressure was obtained, the efficiency being the ratio of the useful work done on the air to the theoretically available energy in the steam supplied to the turbine. Such a high efficiency is impossible in the present state of the turbo-compressor, if indeed it ever will be attainable.

The efficiency of a turbo-compressor has been shown to be less the higher the pressure*, but great difficulties present themselves in the making of actual tests, mainly on account of the uncertainty as to the volume of air delivered.

The figures in Table XXXV. give the results of tests on a fan driven by a Rateau steam turbine. This, however, hardly comes under the same heading as a turbo-compressor.

TABLE XXXV.

Orifice, Sq. In.	Lbs. per Sq. In.		Air Pressure Feet of Water.	Steam per Hour.	H.P.		Net Efficiency.
	Admission	Vacuum.			Theory.	Actual.	
19·1	113·8	4·53	10·60	3,037	260·4	99·0	0·38
	128	4·80	11·61	3,403	299·5	112·6	0·38
22·3	113·8	4 92	9·00	3,037	256·0	93·2	0·36
	128	5·32	9·90	3,403	289·6	106·5	0·37
	142·2	5·68	11·00	3,760	327·2	121·9	0·37
27·8	113·8	4·44	7·55	3,037	261·8	91·1	0·35
	128	4·80	8·25	3,403	299·5	102·5	0·34

Gas Turbines.—Much of what has been said previously as regards steam turbines will apply also to gas turbines.

On the other hand, the gas turbine presents many obstacles which do not confront the designer of a steam turbine. Most of these difficulties are due to the high temperatures which accompany the combustion of gaseous fuel. These difficulties we may divide into two classes ; those belonging to the turbine proper, and those which occur in the combustion apparatus. In addition, there are other difficulties, notably those of compressing the gas and air previous to combustion.

* "A New Work Diagram for Gases," by Frank Foster, in the "Engineer," December 1st, 1905. (See also Chapter XIII.)

This compression must take place outside of the turbine proper. Naturally, the designer's first idea is to use a compressor, or compressors, of the turbine type. As we have seen, however, the efficiency of such a compressor is very low; probably not above 50 per cent., if, indeed, it exceeds 40 per cent. This almost debars the turbo-compressor. For instance, if the theoretical works given out by the turbine and absorbed by the compressor are 100 and 24 respectively, and if their efficiencies are 0·6 and 0·4 respectively, then their actual works will be 60 and 60. That is, the compressor takes all the power the turbine can supply. Nor is the ratio of the (theoretical) compression work to the expansion (turbine) work of the last example excessive. Indeed, rather the reverse is true. Practically, then, the turbo-compressor may be left out of account.

Even with a theoretically perfect compressor it does not follow that a very good efficiency will be obtained. According to Mr. Dugald Clerk, a gas turbine constructed by Mr. F. W. Lanchester, using the exhaust gases from a petrol engine, did actually rotate, but gave no power, even though it had no gas to compress. Mr. Sanford A. Moss, in the United States, constructed a gas turbine of the De Laval type, which actually gave 3 b.h.p. at 19,000 revs. per minute. The air for combustion was supplied from an independent source, and the theoretical work spent on its compression was 4 h.p., so that the net output of the turbine was negative.

The elementary thermodynamics of the gas turbine contains nothing new,* the difficulties to be surmounted are entirely of a physical and chemical nature. In Table XXXVI., which is due to Mr. S. A. Moss, we have some data as to the efficiencies of various gas-turbine cycles. In all cases the heat of combustion is received at constant pressure, and the exhaust takes place at constant pressure. Cases I. and II. refer to the theoretically-perfect engine, Cases III. and IV. to an engine with losses. In Case IV. water is introduced into the gases to reduce their temperature. The turbine is assumed to give 70 per cent. of the theoretical work; the compressor takes 20 per cent. excess of work (i.e., efficiency of 83 per cent.), and the

* See paper on "A Scientific Investigation into the Possibilities of Gas Turbines," by R. M. Neilson, Inst. Mech. Engineers. 1904.

regenerator has an efficiency of 60 per cent. These figures are certainly the upper limit of possible results in practice.

A gas turbine working on the ordinary Otto cycle has been suggested, the chief objections being the fluctuation of the initial pressure and the necessity for repeated explosions with complicated valve arrangements. Fig.

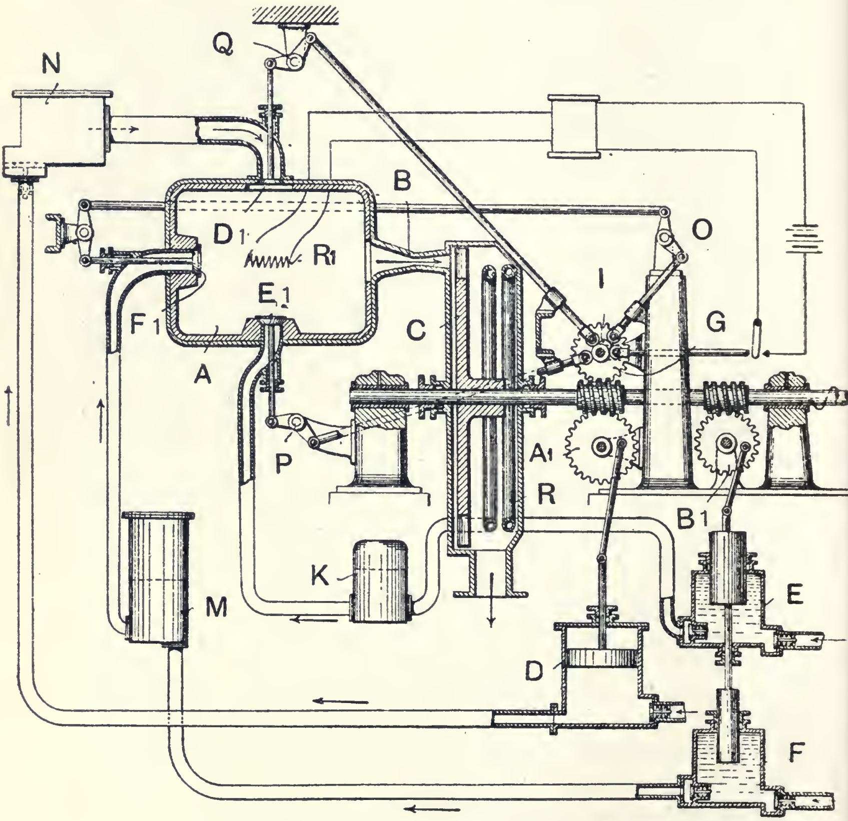


FIG. 196.—ZOELLY EXPLOSION GAS TURBINE.

196 illustrates the suggested arrangement of the Zoelly explosion gas turbine. Air is admitted first into the explosion chamber, and then gas or oil, the air being supposed to act as a shield against back firing. The charge is exploded and when the maximum pressure has been reached, water, previously heated

by being passed through a coil in the turbine-wheel chamber, is injected to reduce the temperature and pressure of the gas. The mixture of vapour and gas is then expanded in the nozzle B, and discharged, at high velocity but relatively low temperature and pressure, against the buckets of a De Laval type of turbine.

The troubles in the combustion chamber of a gas turbine are not fully understood as yet. One serious trouble is the occasional missfire of some of the incoming charge, followed as a rule by a violent explosion. The difficulty of procuring smokeless combustion is another.

It would seem that both these troubles might be mitigated, if not entirely removed, by maintaining the combustion chamber at a high temperature. Any cooling water should be injected into the gases after they have left the chamber. In view of the observed results of ignition at various points in the cycle of an ordinary reciprocating gas engine, it would seem likely that a periodic lowering of the pressure in the combustion chamber would assist the ignition. It is well known that a gas-engine charge explodes more readily during the expansion stroke than the compression stroke.

The ordinary reciprocating gas engine is possible because the mean cylinder temperature is vastly lower than the maximum. The same condition might be attained in the gas turbine by the alternate expansion in the turbine of the working fluid and previously-cooled air, the expansion of which would produce quite low temperatures. In order to prevent the rapid oxidation of the nozzles and blades under such conditions, these alternations would have to be very rapid, for the blades would get so hot whilst in contact with hot gases for any appreciable period that they would literally burn up when free oxygen was presented to them. Such an arrangement would also permit of a high-combustion temperature subject to the periodic fluctuations (small) suggested previously. The mechanical efficiency would be low. It is doubtful if continuous temperatures exceeding 1,000° Fah. are allowable in a gas turbine, but by the use of some cooling

arrangements (whether internal or external) the temperatures attained by the gases might exceed this value.

The high temperatures in a gas turbine would also give rise to grave difficulties in construction, so as to allow for the proper relative expansions of the different parts.

Table XXXVI.

Brayton Cycle Calculations.

	Maximum Pressure, Pounds Per Square Inch Gauge.	Heat Added, B.T.U. Per Pound of Working Substance.	Pounds of Water Per Pound of Working Substance.	Maximum Temperature Degrees Fah. for a Perfect Gas Only.	Final Temperature Degrees Fah. for a Perfect Gas Only.	Ratio of Compressor Power to Net Power.	Velocity of Impulse Wheel. Feet Per Minute.	Net Thermodynamic Efficiency.
	p_1	Q	x	t_2	t_3	r	V	e
Case I.	90	1,000	0	4,665	2,435	·22	115,300	·43
	90	250	0	1,505	652	·87	71,680	·43
Adiabatic compression.	195	1,000	0	4,860	2,008	·27	131,200	·54
Perfect machine with	195	250	0	1,710	546	1·07	83,740	·54
no losses.	495	1,000	0	5,190	1,562	·34	147,600	·64
	495	250	0	2,040	434	1·38	98,300	·64
Case II.	90	1,000	0	9,239	5,039	·08	159,100	·93
	90	250	0	1,965	915	·39	79,570	·72
Isothermal compression	195	1,000	0	7,376	3,176	·11	159,100	·91
with regenerator.	195	250	0	1,498	448	·61	79,570	·62
Perfect machine with	495	1,000	0	6,087	1,887	·15	159,100	·87
no losses.	495	250	0	1,176	126	1·03	79,570	·49
Case III.	56	333	0	2,147	1,200	·74	75,400	·27
	82	392	0	2,398	1,200	·68	84,800	·30
Isothermal compression	61	445	0	2,867	1,619	·50	86,485	·31
with regenerator. Actual	98	545	0	3,233	1,619	·47	99,960	·35
machine with assumed	33	445	0	3,179	2,139	·40	78,155	·28
losses. Air excess.	61	569	0	3,699	2,139	·36	96,775	·33
Case IV.								
Isothermal compression	79	625	·37	1,275	600	·67	75,000	·13
with regen. Assumed	47	638	·36	1,680	1,000	·43	75,000	·15
losses. Cooling water.								

There is no hope that a gas turbine (working between the same pressure limits) can beat the reciprocator so far as

efficiency goes. Its chances of success depend more on small size and simplicity, neither of which seem to be within sight, and indeed, when used in conjunction with a reciprocating compressor and a separate combustion chamber, the turbine seems to be the more complex of the two.

Chapter XIII.

DIAGRAMS AND CALCULATIONS.

Formulæ for the Indicated Horse-power of a Turbine.—Many people desire some means of computing the maximum indicated horse-power which a turbine will give, from its dimensions. The formulæ suggested below are not perfect, but, especially when their constants have been verified and amended by experience, they should prove of use.

The theoretical work done between two pressures per pound of steam is

$$T = \frac{p_1 u_1 - p_2 u_2}{1-m}$$

the expansion curve having the equation

$$p^m u = \text{constant}.$$

The actual work done per pound of steam is

$$t = \frac{k\,(p_1 u_1 - p_2 u_2)}{1-m}.$$

Now if

w = pounds of steam per second,
A = cross-section of annular opening taken perpendicular to shaft,
v = axial velocity of steam,

Then

$wu = Av.$

Hence the total work done per second is

$$W = \frac{k\,(p_1 v_1 A_1 - p_2 v_2 A_2)}{1-m}.$$

The values of p and A are of course known, but there is a difficulty in correctly estimating the values of v. If the blade heights were everywhere proportioned so as to maintain a constant axial velocity for the steam—within any one section—we could immediately determine v from the blade angles and peripheral speed. Usually this is not done, and hence at the commencement of a

section the velocity as calculated from the blade angles would be too large, and too small, at the end of a section.

In some impulse turbines the passage cross-sections do approximate to the theoretical, and hence we can rely

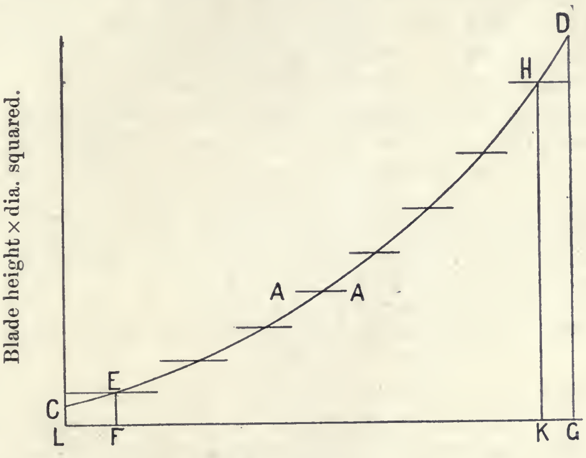


FIG. 197.—APPROXIMATE METHOD OF CALCULATING VELOCITY CORRECTIONS.

upon our calculated values for v. In some reaction turbines the net passage cross-section measured at the middle of the blades is adjusted so as to maintain an approximately constant axial velocity. In these latter cases, however, it would seem that this axial velocity—midway between inlet and outlet edges of the blades—is not that calculated from the blade angles, but is a mean between the inlet and outlet absolute velocities of the steam, and is generally from $1 \cdot 5$ to 2 times the axial velocity, as calculated from the blade angles. Whatever its value it can be used in our formula, provided that we remember to make the value of A equal to the *net cross-section* midway between the inlet and outlet edges in this case.

Where the spacing of the blades is not varied in this manner, or where—as is often the case—we only know the annular openings between rotor and casing, we must attempt to estimate the correct values for v. One method is as follows: Referring to Fig. 197, we measure the number of rows of blades along the horizontal axis, and

the heights of the lines on the diagram we make equal to the products of the blade heights, and the squares of the mean diameters of the blade circles in the respective sub-sections. Thus the short horizontal line A A is obtained by multiplying the blade height of a certain sub-section—really the radial distance between casing and rotor in a reaction turbine—by the square of the diameter. We then draw the curve C D through the middle points of these short horizontal lines, this curve being then approximately a curve of steam volume. Then evidently the ratio of the actual to the calculated axial velocity—referred to total annular opening between rotor and casing—at the entrance to the turbine is as C L is to E F, whilst the ratio of the actual to the calculated for the last row of blades is as D G is to H K.

For instance, suppose we have

$$\begin{aligned} CL &= 0{\cdot}85\, EF \\ DG &= 1{\cdot}3\ \ HK \end{aligned}$$

and the calculated velocities are respectively 180ft. and 450ft. per second, then the actual values will be

$$\begin{aligned} v_1 &= 0{\cdot}85 \times 180 = 153 \\ v_2 &= 1{\cdot}3 \ \ \times 450 = 585 \end{aligned}$$

The value of m usually varies between 0·91 and 0·92, the higher values being as a rule for the higher initial pressures, and the smaller for less efficient turbines.

The value of k usually varies between 0·60 and 0·70, although it may be either less or more. The higher values are for the larger and more efficient turbines.

To reduce our formula to horse-power when the pressures are in pounds per square inch, areas in square inches, and velocities in feet per second, we have

$$\begin{aligned} I.H.P. &= \frac{k\,(p_1\, v_1\, A_1 - p_2\, v_2\, A_2)}{550\,(1 - m)} \\ &= K\,(p_1\, v_1\, A_1 - p_2\, v_2\, A_2). \quad . \quad . \quad (1). \end{aligned}$$

The values of K will lie between 0·012 and 0·016, with an average value of about 0·0135.

Suppose, for instance, that in a turbine working between the pressures of 150lbs. and 1·5lbs. absolute, the total annular passage cross-sections at admission and exhaust are 50 sq. in. and 900 sq. in. We estimate the

values of the axial velocities at these points to be 153ft. and 585ft. per second. Then the indicated horse-power will be about

$$0 \cdot 0135\,[1{,}146{,}000 - 790{,}000] = 4{,}800$$

A much simpler formula based on the average steam consumption of turbines would be

$$I.H.P. = 0 \cdot 0043\, p_1\, A_1\, v_1 \quad \ldots \ldots \quad (2).$$

If the turbine is running non-condensing, this becomes approximately

$$I.H.P. = 0 \cdot 0026\, p_1\, A_1\, v_1 \quad \ldots \ldots \quad (3).$$

As before, there is frequently some trouble in estimating the correct value for the velocity. In order to eliminate this source of error the formula should—for a reaction turbine—take into account the proportion of the rows in the first section which have the same height as the first row.

If a = number of rows (on spindle) in the first sub-section of constant blade height,

b = total rows in first section (of constant diameter), then formula 2 becomes approximately

$$I.H.P. = 0\ 0043\, p_1\, A_1\, v_1 \left(\frac{2\,b - a}{2\,b}\right) \quad \ldots \quad (4),$$

and a similar factor is added for formula 3.

For instance, if the turbine we have just been considering had 30 rows in the first section and of these the first 10 were of uniform height we see that the indicated horse-power is 4,830.

For small and inefficient turbines the indicated horse-power will be from 5 to 35 per cent. less than the values calculated from the above formulæ, and for the best turbines will be somewhat higher.

Formula 4 is probably the best formula for ordinary use, as it can be readily calculated from measurements on the turbine, v_1 being the axial velocity in feet per second as calculated from the velocity diagram and is equal to

$$\frac{V \sin \alpha \sin \beta}{\sin(\alpha - \beta)}$$

V being the peripheral speed of the blades, α and β the blade angles. A, of course, is the area of the

annulus between rotor and casing in square inches. Pressures are absolute.

Generally speaking, the brake horse-power is of more consequence than the indicated, and to obtain it we multiply the constants in our formulæ by about 0·95. This reduces them to 0·0128, 0·0041, and 0·0025 respectively.

The method of calculating the velocity of the steam from the blade shapes assumes that the number of rows of blades has been correctly designed. If this has not been done then some other means of estimating v must be adopted.

Heat Diagram.—The diagram about to be described will be found useful for steam turbine and general steam problems. If we consider what occurs in the cylinder of a steam turbine, we see that part of the kinetic energy from which the turbine derives its power to do work is converted into heat and largely wasted, thus reducing the efficiency of the turbine. The heat thus generated increases the volume of the steam either by evaporating moisture or by superheating. Clearly, then, the areas of the ordinary pressure-volume and entropy diagrams will be increased beyond those which they would have for adiabatic expansion, so that these areas cannot now represent the actual quantities of work done.

At the expense of some extra labour, it is possible to calculate the actual work done in the turbine from the pressure-volume or entropy diagram.

The author's heat diagram enables the quantity of work done, the losses, velocities, pressures, volumes, dryness fractions, and temperatures to be read off directly from the diagram. The diagram is sketched (not to scale) in Fig. 198, temperatures being measured along the axis O T and heat quantities along the axis O H. We know that the total heat increases uniformly* with the temperature T, so that we can draw in the line A B to represent the relation between the temperature and total heat of 1lb. of saturated steam. A B may be called

* The increase is not strictly proportional. Still it is sufficiently so to be taken as proportional for most practical purposes, and in any case the description of the diagram given here will suffice.

the "saturation" or "steam" line. Similarly, we can draw in the "water" line C D to represent the total heat of water at all temperatures. We know that the total heat of steam at t degrees Fahrenheit is given by

$$H = 1082 + 0{\cdot}305\ t$$

and the total heat of water (usually called sensible heat) by

$$h = t - 32.$$

Similarly, we can draw in the total heat lines for any given dryness fraction. If E K is such a line and x the percentage dryness, then the horizontal distance of E K (at any point) from C D is to the horizontal distance from A B to C D as x is to 100.

For saturated steam the pressure is fixed by the temperature, so that constant-pressure lines will be horizontal. When we know the specific heat of superheated steam with certainty, we can draw in the constant-pressure lines for superheated steam also. At present we do not know the specific heat, so that perhaps the best thing to do is to assume it constant, in which case, the constant-pressure lines will be straight, L P being intended to represent such a constant-pressure line.

We can also draw in a series of adiabatic lines such as F G.

If L = latent heat.
T = absolute temperature (Fahrenheit degrees)
= 461 + t.
s = dryness fraction.
H = total heat.

Let the suffix 1 refer to the initial—or some known—condition of the steam, then we have

$$s\frac{L}{T} = \log\frac{T_1}{T} + s_1\frac{L_1}{T_1}$$

and

$$H = T - 493 + s\left(\frac{L}{T}\right)T$$

$$= T - 493 + T\left(\log\frac{T_1}{T} + s_1\frac{L_1}{T_1}\right)$$

all logarithms being hyperbolic (equal to 2·3026 times the common logarithm).

The above equation is of the form

$$H = m\ T + c + T\log\frac{T_1}{T}$$

and deviates from a straight line by the last term in the equation. For many purposes it is sufficiently accurate to draw in a mean straight line, in which case the last term disappears from the equation, and we have

$$m = 1 + s_1 \left(\frac{L_1}{T_1}\right) + \log \frac{T_1}{T_2}$$

where T_2 is some temperature during the expansion—usually the final temperature, but sometimes a mean temperature according as to whether the final or average

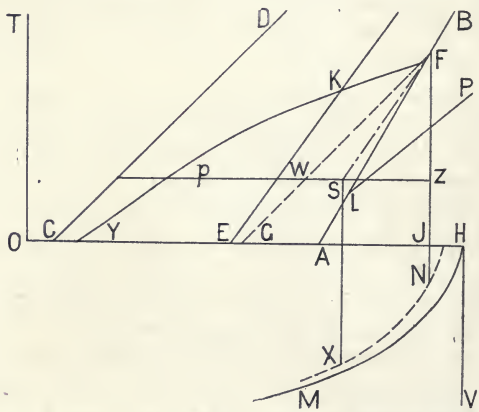


FIG. 198.—HEAT DIAGRAM FOR STEAM OR OTHER VAPOUR.

conditions are the more important in the particular problem under consideration.

Vertical lines, such as F J, will be lines of constant heat or "throttling" lines. They represent the changes which occur when the steam in expanding does no external work, and does not change its velocity. If we draw adiabatics from various points on such a throttling line to a common back (condenser) pressure we see immediately how throttling the steam reduces the quantity of available work in it.

We can easily draw in constant-volume lines for saturated steam. Let

U = volume of 1lb. of dry saturated steam.

t = temperature.

u = actual volume.

h = water heat.

w = volume of 1lb. of water.

Then the total heat at this temperature is

$$H = h + \frac{(u - w)\,L}{U - w}$$

$$= t - 32 + \frac{(u - w)\,L}{U - w}.$$

This gives us the constant-volume line Y K.

Since for a free expansion the increase in kinetic energy is equal to the decrease in total heat, we can say that

$$\frac{v_2^2 - v_1^2}{2\,g} = H_1 - H_2.$$

Hence draw an axis H V along which to measure velocities. Then we can draw in a series of parabolas (all from the same template) which satisfy the equation

$$v^2 = 2\,g\,H.$$

In this case H is supposed to be in foot-pounds. H M and N X are two such lines. By their use we can at once read off the change in velocity produced in any given free expansion.

Use of the Heat Diagram.—Suppose that in Fig. 199 F S is an expansion line, F W an adiabatic through the initial position F, and F Z a throttling line, or line of constant heat. Then at the pressure W S we know that the heat in the steam is represented S and is less than the initial heat in the steam by S Z. Consequently S Z must be the amount of heat converted into either kinetic energy or useful work. If the expansion had been adiabatic, W Z would represent this work done, so that the ratio of S Z to W Z gives us the efficiency of the expansion.

So far we have neglected the effect of a steam jacket or of radiation. Since radiation will decrease the heat remaining in the steam, and a steam jacket will increase it, then from S Z we subtract the loss by radiation in order to give us the actual work done, and to S Z we add the heat received from the jacket in order to obtain the actual work done. The radiation loss can seldom be estimated with accuracy, but fortunately this is not of much importance. The jacket heat can generally be

estimated from the condensation in the jackets. For most problems the scale of heat may commence at about H = 850 B.Th.U.

Suppose that the expansion is a free expansion, the initial velocity being H N. Draw the velocity curve N M R through the point N as in Fig. 199. Then K M will be the velocity at the end of the expansion and L R the velocity which would have been acquired if the expansion had been adiabatic.

The question arises as to how we are to determine the real expansion line F S. For a reciprocating engine it can, of course, be obtained directly from the indicator diagram,

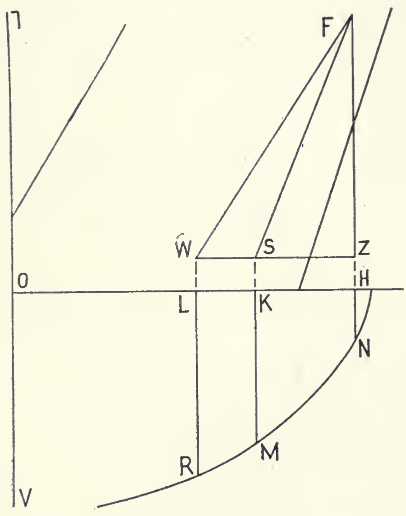


FIG. 199.—HEAT DIAGRAM SHOWING VELOCITY GENERATED DURING EXPANSION WITH LOSSES.

and for a turbine it may be approximated to by taking samples of steam at different points, and determining their dryness fractions, pressures, and temperatures. This method is not very accurate, it being hardly possible to obtain fair samples of steam under the conditions existing inside a turbine.

For most purposes we have to be content with estimating the expansion line from the adiabatic. For instance, suppose steam initially dry at F (Fig. 198) to

expand to the pressure p, and that at any point in the expansion we assume that 30 per cent. say of the available heat has been used up in friction and eddies. This heat will have been spent in drying the steam. If the pressure line p cut the adiabatic G F in W and the throttling line F J in Z, make W S equal to 30 per cent. of W Z. Join F S. Then F S is the actual expansion line and S Z is the heat converted into useful work or kinetic energy, as the case may be.

Suppose this heat to have been converted into kinetic energy. Project down from Z on to one of the velocity curves such that the intersection J N is equal to the velocity at the commencement of the expansion. Then the projection A X *on the same velocity curve* from S gives us the final velocity at the end of the expansion. If, as in most turbines, this expansion has been in stages, the kinetic energy will have been absorbed by the moving blades approximately at the same time as it was produced, and hence A X will not represent the final velocity. If the expansion line represented only one set of fixed blades then the velocity would be represented by A X.

For most turbine problems we can assume that there is a loss of 3 or 4 per cent. by conduction and radiation from the turbine. This can be allowed for by setting back the point Z towards W by from 3 to 4 per cent. of Z W. Then we can assume that the conversion of kinetic energy into heat represents from 25 to 35 or even 45 per cent. of the available heat W Z, the lower figures being for large turbines using superheated steam and well designed. This is represented by W S, and the rest of the heat is spent in doing work—some of it in turning the rotor against the resistance of the steam and the bearings—and in rejecting kinetic energy into the exhaust pipe. For multi-stage turbines this latter is small, not often more than 6 or 8 per cent. of W Z, and generally only about half this amount in Parsons turbines.

If this construction is adopted it is important to notice that the expansion line $F\ S$ is not now the true expansion line. If we desire to obtain the true expansion line, and with it the steam volumes and dryness fractions, we must set W and not Z back by an amount equal to the radiation loss, as such loss must cause condensation.

In this case—Z remaining untouched—the work done is equal to $S\ Z$, less twice the radiation loss.

With regard to the percentage conversion of kinetic energy into heat, data are too scanty to enable us to be very dogmatic at present, but we shall probably be correct in assuming that the low-pressure section is more efficient than the high-pressure, the difference being most marked in small turbines, and less when using fairly highly superheated steam.

Volumes on Heat Diagram.—We have seen how constant-volume lines can be drawn in on the diagram from which we can readily read off the volume. These lines are a matter of considerable labour to draw, so that

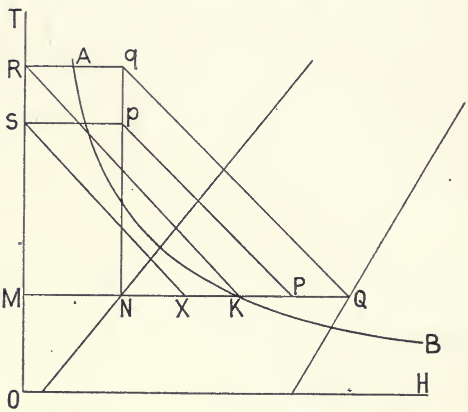


FIG. 200.—GEOMETRICAL CONSTRUCTION FOR OBTAINING VOLUMES ON THE HEAT DIAGRAM.

it is generally advisable to determine the volumes by one of two methods about to be described, usually the latter method.

The first method is a simple geometrical one, and is indicated in Fig. 200. We want to know the steam volume at the point P. Draw the constant-temperature line N P Q, cutting the curve A B in K; A B being a curve showing the volume of 1lb. of dry saturated steam at all temperatures. Draw N p q parallel to O T and make

$$M\ S = Np = N\ P.$$
$$M\ R = Nq = N\ Q.$$

Join R K, and draw S X parallel to R K. Then M X is the steam volume at P. There are several objections to the above method, in particular the fact that the water line is seldom on the chart at all—in order to secure a more open though more limited scale for the quantities of heat.

The method preferred by the author is as follows :—

Let L = latent heat at this pressure.
u = actual volume of steam and moisture.
U = volume of 1lb. of dry steam.
w = volume of 1lb. of water = 0·016 cub. ft.
H_1 = total heat dry saturated steam.
H_2 = total heat of steam in condition represented by P.

The increase in volume during evaporation from water to the condition of dry saturated steam is $U - w$, and the (increase in) volume per unit of heat is

$$Q = \frac{U - w}{L}.$$

Now, it takes $H_1 - H_2$ units of heat to completely evaporate the moisture in the steam, so that the volumetric increase due to such extra heat would be

$$Q\ (H_1 - H_2),$$

making the volume equal to U, so that evidently the volume at P is

$$u = U - Q\ (H_1 - H_2).$$

The author has calculated Q, and hence, $H_1 - H_2$ being measured directly off the diagram, u is readily calculable. (Table LXVI. in appendix.)

Fig. 201 is a scale drawing of the heat diagram. The horizontal lines are constant-pressure lines for wet steam. The single slightly-inclined (from the vertical) line is the steam line, representing the total heat of dry saturated steam. The approximately parallel inclined lines are adiabatics. The figures marked on these lines indicate the pressures at which the steam is dry and saturated.

In order to assist readers who wish to construct a large scale heat diagram, Table XXXVII. has been compiled. Thus in the vertical column under the pressure 60 and opposite (horizontally) the pressure 130, we read 1,129·5.

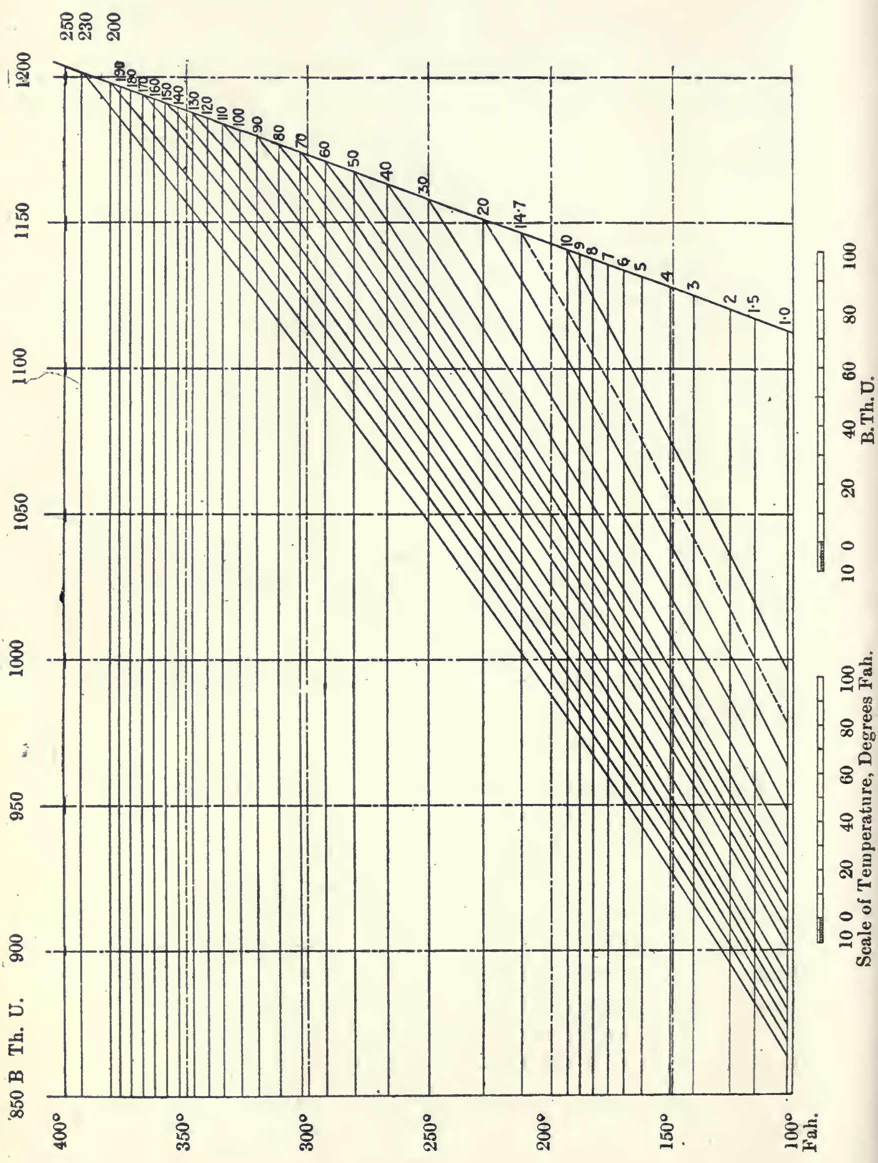


FIG. 201.—FOSTER'S HEAT DIAGRAM FOR STEAM.

This means that the adiabatic for dry steam at 130lbs. absolute has a total heat of 1,129·5 when at a pressure of 60lbs.

TABLE XXXVII.

Initially dry at	Total heat during expansion at lbs. square inch.						
	150.	100	60.	30.	10.	4.	1.
230	1,166·0	1,133·0	1,094·8	1,047·0	980·0	927·2	863·0
200	1,175·5	1,143·0	1,104·5	1,056·0	987·5	934·0	868·5
170	1,185·5	1,161·5	1,113·5	1,064·0	995·0	940·7	874·0
150	1,191·2	1,169·1	1,121·7	1,071·6	1,001·5	946·7	879·0
130		1,177·2	1,129·5	1,079·0	1,005·9	953·0	884·6
110			1,137·8	1,087·0	1,017·0	959·8	890·5
90			1,148·0	1,097·0	1,026·5	970·0	900·5
80			1,155·8	1,104·0	1,033·0	975·6	906·5
70			1,163·8	1,113·0	1,041·2	982·5	913·0
60			1,171·2	1,121·0	1,047·7	989·0	919·0
50				1,129·0	1,055·5	996·0	925·5
40				1,143·5	1,067·5	1,008·0	935·5
30				1,158·3	1,081·5	1,021·6	947·5
20					1,103·5	1,039·0	962·5
14·7					1,120·0	1,056·5	977·7
10					1,140·9	1,075·0	995·7

Mollier's Diagram.—Professor Mollier has devised a modified entropy diagram for use in studying steam problems. Entropy is measured along one axis, and total heat along the other. A constant-pressure line on this diagram will show the relation between the total heat and entropy during evaporation and superheating. The saturated steam line shows the relation between the entropy and the total heat of dry saturated steam.

One serious objection to the diagram is the fact that only adiabatic lines—lines of constant entropy—are straight. A great deal of labour is required in the construction of one of these diagrams, and it is more difficult to use than the heat diagram previously described.

Many-stage Impulse Turbine.—In Fig. 202 $F\ S$ is the expansion line for one section of the turbine in which the diameter is constant, and hence in which the work done in each stage is the same; $S\ F$ having been determined by the method previously explained. We have previously decided the blade angles and speed, and hence also the (indicated) work done in each stage. If this work is W,

mark off (see figure) $H_1 H_2$, $H_2 H_3$, &c., each equal (in terms of heat units) to W. Project up on to the expansion line and the intersections give us the steam pressures, temperatures, and volumes.

This construction will also give us the number of stages in this section. Where two or more different diameters are used in the turbine the method of procedure is the same, except that after the steam has attained a certain volume—and therefore some known temperature—the work done in each stage, W, will be increased in the ratio of the squares of the diameters. The number of stages in each section is, however, more readily determined by direct calculation. For instance, if the inlet and outlet pressures of a section are 135lbs. and 59lbs. per square inch absolute, the available work between these limits (assuming the initial steam to be just dry) is 64·8 British thermal units. Of these assume that 45·3 or 70 per cent. are converted into indicated work. If the peripheral

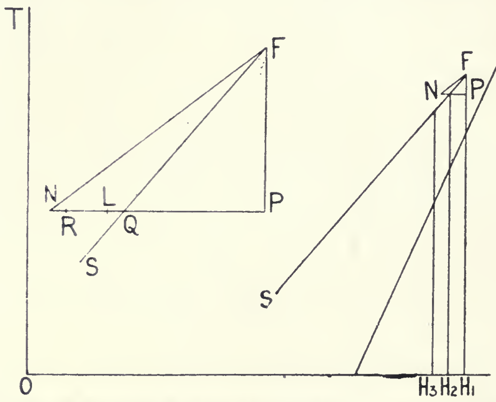


FIG. 202.—MANY-STAGE IMPULSE TURBINE. HEAT DIAGRAM.

velocity of the steam is 200ft. per second and the blade angles are: Nozzle outlet 35 deg. and moving-blade angles both 48 deg., the work done in each stage will be

$$\frac{200}{32 \cdot 2}(2 \times 320 \cos 48)$$

$$= 2{,}745 \text{ foot-pounds per pound per second,}$$
$$= 3 \cdot 53 \text{ B.Th.U.}$$

Hence we should have 13 stages in this section. The passage cross-sections are readily calculated when we

know how much steam the turbine is expected to pass per second. Thus if the *maximum* capacity of the turbine—without opening the by-pass—is 1,200 kw. at an estimated steam consumption of 21lbs. per kw. hour, the turbine will pass 7lbs. per second. From the peripheral speed—200ft. per second—of the blades and their angles we see that the axial velocity of the steam is 238ft. per second Hence if the volume of the steam at a certain point along the turbine is u cubic feet per pound, the area of the passages in the diaphragms, measured in a plane perpendicular to the axis, will be

$$A = \frac{7\,u}{238} \text{ square feet.}$$

Thus if at the entrance to the first set of fixed blades the pressure is 130lbs. absolute and the volume 3·38 cubic feet per pound, the area of the passage cross-section will be 14·3 square inches. If the revolutions per minute are 1,800 the diameter will be 25·5in., with a circumference of 80in. We might then group the nozzles—passage openings—into four equidistant groups, each 4in. long by 0·893in. radial depth, or one group, which would perhaps be better.

Actually it is unusual to calculate these passage areas for the inlet and outlet of each set of fixed blades, because we should find that the inlet depth would be a little greater than the outlet depth and, except possibly where there were only a very few stages, the cost of manufacture when these trifling differences were attended to would be prohibitive. Consequently the mean steam volume during the expansion in a set of blades may be used in the calculations, although it is undoubtedly much better practice to base the calculations on the volume at outlet. In most multi-stage turbines the difference, especially in view of the uncertainty of the actual conditions, is too trifling to be a matter of consequence except at the exhaust end.

Towards the low-pressure end of the turbine, where full peripheral admission is employed, it is common practice to make several consecutive stages identical in size, so that the passage cross-section is determined for the average steam volume in these stages, and most of the velocities are somewhat incorrect, unless the blade angles are varied from stage to stage.

Curtis Turbine.—The first thing is to draw a velocity diagram for the steam,* making allowance for the losses from point to point in the stage considered—all the stages are commonly identical so far as velocities go. Exactly what these losses should be it is impossible to say, but the following may serve as a guide for a turbine with two sets of moving blades per stage :—

	Losses per cent.
In nozzles	5 to 8
Nozzles and 1st blades	4 to 6
In 1st blades	5 to 8
1st and guide blades	3 to 5
In guides	4 to 6
Guides and 2nd blades	2 to 4
2nd (moving) blades	2 to 3
In exhaust	3 to 6
Total losses of kinetic energy	28 to 46

After allowing from 4 to 8 per cent. for leakage and radiation from the turbine, the remainder will represent the indicated work of the steam, and should equal the indicated work calculated from the velocity diagram after allowing for the losses. It may be as well to point out that the above losses refer to kinetic energy, and, moreover, of the total kinetic energy. For instance, suppose that the total available kinetic energy is 100, and that the kinetic energies entering the first and second sets of moving blades are 90 and 25 respectively. Then, if the losses of these two sets of blades are 5 and 2 per cent., the actual losses will be 5 and 2, not 5 per cent. of 90 and 2 per cent. of 25.

The figures for the losses given in the preceding table are only given because there are no better. The lower losses may be taken as those in a large turbine; the others in a smaller turbine. It would appear, too, that the losses in the high-pressure section are greater than those in the low-pressure section.

The steam volumes are easily determined. The volume produced by an adiabatic expansion is, say, 10 cub. ft. There have been certain losses of kinetic energy—say 15 B.Th.U.—previous to the arrival in the guides There has also been a loss of, say, 2 B.Th.U. by radiation and

* See Fig. 79, page 87.

leakage, so that 13 B.Th.U. have been spent in increasing the volume of the steam by an amount which is easily calculated. All the above can be determined by the heat diagram. We now know the volume and velocity of the steam, so that the passage cross-sections can be easily calculated.

Reaction Turbine.—The method of procedure is practically the same as for a multi-stage impulse turbine. We first determine the actual expansion line on the heat diagram. Thus, referring back to Fig. 202, the expansion line is $F\ S$. Divide this up as for the impulse turbine by projecting up from the points $H_1 H_2$, &c. Since half the expansion takes place in the fixed, and half in the moving blades, we ought to split the above expansions into two, were it not for the fact that it is common practice to give several consecutive stages the same passage cross-section. In this case we calculate the passages from the average steam volume. The reason for this is of course merely to cheapen the cost of construction.

Design of 3,000 kw. Parsons Turbine.—In order to illustrate more definitely the design of a reaction turbine, let us assume an example as follows:—

Rated load = 3,000 kw. at 50 cycles.
R.P.M. = 1,500
Stop-valve pressure = 180lbs. absolute.
Exhaust pressure = 1·5lbs. abs. (27in.).
Steam dry and saturated.

Theoretical work per pound of steam between the above limits of pressure = 305.

Let us divide the rotor into three main sections having average blade speeds of 135, 220, and 330 feet per second.

The blade angles we will take to be 40 and 62 degrees throughout. Then for a blade speed V and steam velocities v_1 and v_2 (see section on blades, Fig. 81), the work done per pound per second is

$$W = \frac{V}{g}(v_1 \cos 40 + v_2 \cos 62)$$

$$= \frac{V}{g}\left(\frac{V \sin 62 \cos 40}{\sin 22} + \frac{V \sin 40 \cos 62}{\sin 22}\right)$$

$$= 0{\cdot}0812\ V^2 \text{ foot-pounds.}$$

In Table XXXVIII. are given the values of W in foot-pounds and B.Th.U. per stage for the different sections. Certain losses by radiation and leakage have been assumed for each section, and certain (hydraulic) efficiencies from which the indicated works have been calculated. Thus in Section I. we have allowed 4 B.Th.U. for radiation and leakage out of a total of 83. We have left 79 B.Th.U., which, when multiplied by the efficiency of 64·8 per cent. gives us 51·2 B.Th.U. spent in indicated work. Now the average indicated work per stage is 1·9 B.Th.U. Dividing 51·2 by 1·9 we get the number of stages, which is 27.

Table XXXVIII.

	1st Section.	2nd Section.	3rd Section.
Average blade speed, V.	135	220	330
Work per stage, per lb. per sec., ft.-lbs.	1,480	3,930	8,840
" " " " B.Th.U.	1·90	5·05	11·35
Theoretical work, B.Th.U. per lb.	83	109	113
Radiation	4	3	3
Available work	79	106	110
Hydraulic efficiency, per cent.	64·8	66·7	72·2
Indicated work	51·2	70·7	79·5
Stages	27	14	7

The total number of stages is thus only 48, and the overall hydraulic efficiency is 68·2 per cent. In the above we have, however, neglected the bearing and disc frictions. If we assume this at 5 per cent. of the indicated work, the thermal efficiency referred to brake horse-power is 64·7 per cent. (allowing for leakage). If we assume the efficiency of the generator to be 95 per cent., and include the radiation and leakage with the other losses, the net thermal efficiency referred to the power generated is 59·5 per cent. This is perhaps about as good as we can expect.

In general a turbine is designed to take a fair amount of superheat, and perhaps a higher vacuum. These conditions will, of course, increase the number of stages. A decrease in the blade speeds or a suitable change in the blade angles would also lead to an increased number of stages, which, however, is what we want to avoid. The fewer the number of stages the cheaper the turbine. For

a turbine of this size, the above angles are, perhaps, hardly the best.

As regards the blade lengths. In Fig. 203 we have the heat diagram for this case. The line $F\,A\,B\,W$ is the adiabatic and $F\,E\,K\,G$ the actual expansion line; $L\,C$, $N\,D$, and $M\,I$ the radiation losses in the first, first and second, and all three stages respectively. Then $E\,L$, $K\,N$, and $G\,M$ are the indicated works in the first, first and second, and all three stages respectively. $A\,E$, $B\,K$, and $W\,G$ are the internal losses which give rise to a throttling effect on the steam, less the radiation losses.

We now calculate the volumes by the method previously described. For instance, at a pressure of 35lbs. the distance of the actual expansion line $E\,K$ from the dry steam line $F\,X$ is 48 B.Th.U. The quantity Q is 0·01249 cubic feet per B.Th.U. The volume of dry saturated steam is 11·66 cubic feet. Subtract from this the product of Q and 48, and we obtain the actual steam volume at this pressure, namely, 11·06 cubic feet per lb. We do this for a fair number of points spaced at approximately equal temperature intervals, and thus obtain the actual steam volumes at all points in the expansion. In this way Table XXXIX. has been calculated :—

Table XXXIX.

Pressure, lbs. per square inch.	Volume Saturated Steam.	Temp. Fah.	Cubic feet per B.Th.U.	Heat units required to dry steam.	Volume less than dry steam.	Actual volume per lb.
p	U	t	Q	H	$Q\,H$	$u = U - Q\,H$
180	2·49	373	......	0	......	2·49
90	4·80	320	·00539	20·6	0·111	4·69
60·5	6·97	293	·00766	32	0·245	6·725
60·5	6·97	293	·00766	32	0·245	6·725
35	11·66	259	·01249	48	0·6	11·06
21	18·84	230	·01976	62	1·22	17·62
12·4	31·0	203	·0319	76·6	2·45	28·55
12·4	31·0	203	·0319	76·6	2·45	28·55
6	61·14	170	·0614	94	5·77	55·37
3	117·3	141·6	·1154	112	12·5	104·8
1·5	226	114·3	·2175	133	29·0	197·0

We now calculate the mean axial velocity of the steam in each section from the velocity diagram for the blades:—

TABLE XL.

Pressure.	$\frac{u}{v}$, square feet.	$w\,a$ square feet.	Mean diam. in inches.	Mean axial velocity, feet per sec.	Calculated best radial distance, ins.
p	a	A	d	v	r
180	·01216	·241			·535
90	·0229	·453	20·6	204·4	1·01
60·5	·03285	·65			1·444
60·5	·0202	·399			·545
35	·0332	·656	33·6	333	·897
21	·0529	1·045			1·425
12·4	·0855	1·69			2·31
12·4	·057	1·127			1·02
6	·1104	2·19	50·5	500	1·985
3	·210	4·15			3·77
1·5	·394	7·79			7·06

Notes.—The quantity w is the number of pounds of steam passing per second = 19·8.

The radial distance is the radial distance between the surfaces of the drum and casing.

We will suppose that the turbine has to give 3,500 kw. at the generator without opening the by-pass valve. To allow for the thickness of the blade edges, errors, and general contingencies we will make it 3,800 kw. Now, from the known thermal efficiency of the turbine at the generator terminals and the available heat per pound of steam, we see that the turbine takes 18·75lbs. per kilowatt-hour, or 19·8lbs. per second. This enables us to calculate the total steam volumes at different points in the expansion. Dividing these volumes by the respective axial velocities, we obtain the best annular areas of the steam passages between rotor and casing. From these and the mean diameters of the sections we at once have the best blade heights, or rather radial distances between rotor and casing. As was pointed out when discussing blades, it is too expensive a construction to make the actual blade heights equal to these calculated lengths.

TABLE XLI.

Section.	No. of Stages in Sub-section.	Pressure Limits of Sub-section.	Calculated Best Radial Distances.	Actual Radial Distances.	Blade Length in inches.	Radial Clearance in inches.
1 High Pressure	9	180 125	·535 ·72	·61	·58	·03
	8	125 91	·72 ·96	·84	·81	·03
	5	91 74	·96 1·17	1·05	1·02	·03
	5	74 60·5	1·17 1·444	1·30	1·27	·03
2 Inter-mediate	4	60·5 39	·545 ·80	·64	·595	·045
	4	39 25	·80 1·21	·97	·925	·045
	3	25 17	1·21 1·68	1·45	1·405	·045
	3	17 12·4	1·68 2·31	1·96	1·915	·045
3 Low Pressure	3	12·4 4·7	1·02 2·43	1·60	1·55	·05
	2	4·7 2·6	2·43 4·20	3·28	3·23	·05
	2	2·6 1·5	4·20 7·06	5·46	5·41	·05

Note to Table.—Diameters of drum surface in sections, 19·6, 32·3, and 47 inches.

What we do is to divide the stages into sub-sections of constant blade height, that height being equal to the average calculated height for the sub-section.

Table XL. shows the calculated values and Table XLI. the actual values, both with and without allowance for the radial tip clearances of the blades. These radial distances between rotor and casing are also marked on the heat diagram (Fig. 203), which we use to show us the respective pressures at the different stages.

With regard to the use of a constant blade height for several consecutive stages, this will, of course, cause some inaccuracy in the steam velocities. Attempts are

sometimes made to allow for this by varying the spacing of the blades from stage to stage; but this does not appreciably affect the axial steam velocities at inlet and outlet, which are the most important. A better method is to use the same blades and blade heights, but to vary

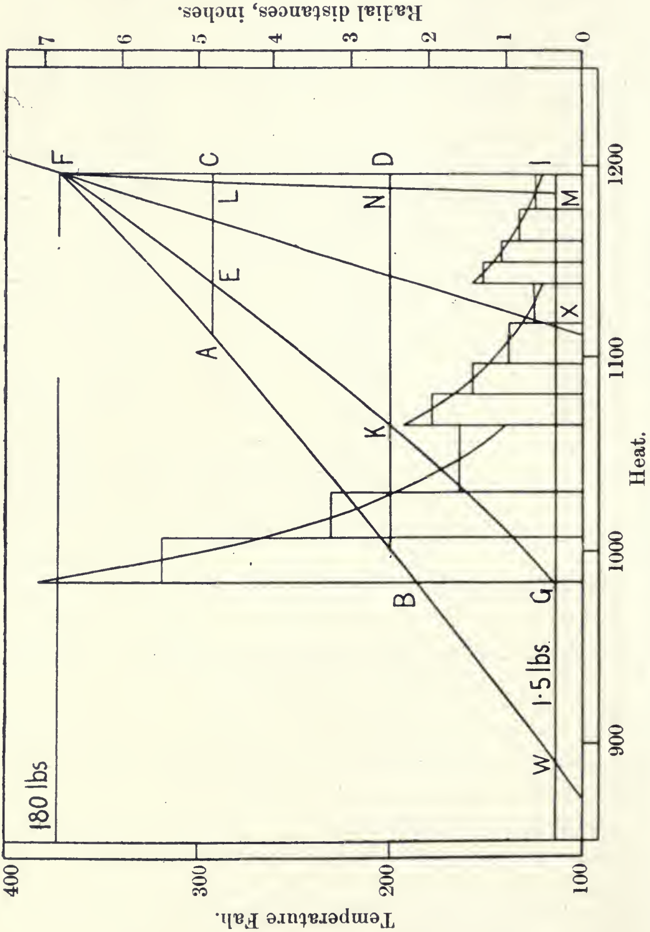


FIG. 203.—HEAT DIAGRAM FOR 3,000 KW. REACTION TURBINE.

the angular positions of the blades relative to the direction of the flow of steam. That is to say, if the normal position of the blades gives inlet and outlet angles of 62 deg. and 40 deg., then the same blades would give

angles of, say, 65 deg. and 37 deg., or 60 deg. and 42 deg. By suitably adjusting these angles the axial velocity of the steam can be varied in proportion to the volume of steam passing, thus allowing of correct blade shapes with a constant blade height. Of course, the works per stage will not now be quite the same in each stage, but if the calculations are based on the mean angles the error will be as small as our other assumptions will require. So far as the author is aware, the above method has not often been tried, but it certainly ought to give good results.

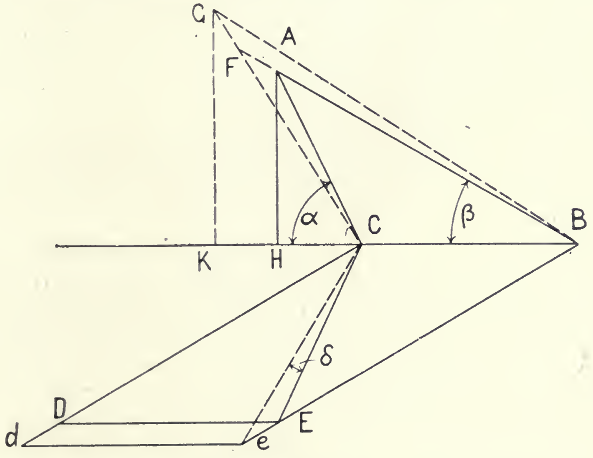


FIG. 204.—VELOCITY DIAGRAMS FOR REACTION TURBINE. CONSTANT BLADE HEIGHTS, VARIABLE ANGLES.

Another, but rather less desirable, method consists in the use of several kinds of blades.

As regards the determination of the angular positions of the blades: Referring to Fig. 204, we have—

$A\ B$ = absolute outlet velocity from first set of fixed blades and inlet velocity for the first set of moving blades.

$A\ C$ = inlet velocity relative to first set of moving blades.

Then, if there were no expansion, $C\ E$ would be the absolute outlet velocity from the first set of moving blades and also the inlet velocity for the second fixed

blades. But there is expansion in the moving blades, and the relative outlet velocity, whilst not altering in direction, increases from $C D$ to $C d$, so that the absolute velocity of inlet to the second fixed blades has increased from $C E$ to $C e$, and reduced its angle α to $\alpha - \delta$.

By geometrical construction we make $C F$ equal to $C e$ and $B F$ equal to $B e$, so that the triangle $C F B$ is equal to the triangle $C e B$. Now, if there were no expansion in the fixed blades (second set) the outlet velocity would be equal to $F B$, but its inclination would be $\beta + \delta$. The expansion, however, increases the velocity to $G B$, which is also the absolute inlet velocity to the second set of moving blades, and for which $G C$ is the relative inlet velocity. We thus see that the *fixed and moving blades for any one stage are exactly similar* in position—that is to say, if the angles for the fixed blades are 60 and 38, the angles for the corresponding moving blades will be the same.

Furthermore, from the geometry of the figure we see that the point G lies on the bisector of the angle $C A B$. Since the blade height is constant, the axial velocities $A K$ and $G H$ must be proportional to the steam volumes at the entrances to the first and second sets of fixed blades. These volumes we determine from our heat diagram. We then construct (Fig. 205) a normal-angled diagram $A B C$ for the mean stage. The length $A K$ for this diagram, and the known steam volume which it represents, gives us the scale of the other perpendiculars, which we can at once draw in. The number of stages in each sub-section should be less at the low-pressure than at the high-pressure end. The above method of varying the blade angles is most useful at the low-pressure end.

Reference to Fig. 203 will assist the reader in following the method of procedure in the determination of the blade lengths. Having calculated the radial distances between rotor (drum surface) and casing, we plot these and obtain the curved lines as shown. This curve for the low-pressure section, for instance, is constructed on the projection of the corresponding expansion line G K as base. We know that there are seven stages in the section. We judge that a good arrangement would be to have three sub-sections with three, two, and two stages

respectively. Each stage absorbs 11·35 B.Th.U. so that the total work done is 34·1 at the end of the first sub-section, 56·8 at the end of the second sub-section, and 79·5 at the end of the last sub-section. In addition, we have to allow for three B.Th.U. as radiation and leakage, making the heats which have disappeared from the steam as 35·1, 58·8, and 82·5 B.Th.U. respectively. Measure these quantities off from the vertical through K, and projecting up on to the expansion line G K the intersections give us the pressures at the entrances and outlets of the respective sub-sections. The intersections of the same upright projection lines with the curve of calculated radial distances enable us to determine the mean radial distances for each sub-section. These are repre-

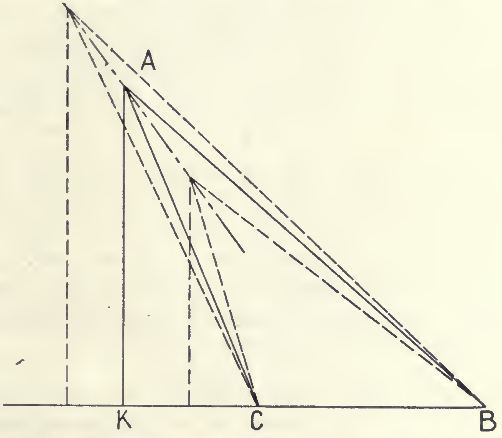


FIG. 205.—BLADE ANGLES FOR REACTION TURBINE. CONSTANT BLADE HEIGHTS.

sented by the stepped horizontal lines on the diagram. From these radial distances we deduct suitable radial clearances, and we have left the effective blade lengths.

With regard to the radiation losses L C, N D, and M I, it should be pointed out that these are pure radiation losses and not leakage losses. A leakage—from the turbine—can be allowed for by adding a certain extra percentage to the steam consumption. These radiation losses, of course, include heat leakage by conduction, but not leakage of steam or water.

A little consideration of the mode of calculation of the blade lengths will show that for small turbines the blades

are likely to be very short at the high-pressure end, and particularly so for a 25-cycle generator, as then the maximum revolutions are only 1,500, and the diameter is therefore very large and the blade lengths proportionately short. Under these circumstances two courses are open to us.

We may reduce the peripheral speed of the blades. For a given r.p.m. the blade lengths are inversely proportional to the squares of peripheral speeds, so that reducing the peripheral speed from 135ft. to 100ft. per second will increase the blade lengths by 82 per cent. To a certain extent this plan is followed in practice. Another method is to adjust the blade angles so that the axial steam velocities are reduced. For instance, a change in the blade angles from 62° and 40° to 52° and 30° reduces the axial steam velocities in the ratio of 1·515 to 1·05, and therefore increases the blade lengths by 44 per cent. If the blade angles are 90° and 30° the increase in the blade lengths is no less than 163 per cent.

One of the most serious objections to the reduction of the peripheral speed in order to obtain reasonable blade lengths, is that the number of stages required goes up inversely as the square of the speeds. Thus a uniform reduction of this speed of 20 per cent. would increase the number of stages in the example we have just worked out from 48 to 69.

End Thrust.—The end thrust for our 3,000 kw. turbine is partly due to static pressure on the steps and end of the drum, partly to pressure on the blades, partly to friction, and partly to the steam velocities differing from the calculated ones. We will assume the two last to neutralise each other.

In calculating this end thrust we shall consider only that portion of the rotor outside a cylinder equal in diameter to the smallest rotor diameter (Fig. 206). The pressures on all areas situated within this cylinder clearly neutralise each other, providing that the shaft has the same diameter at the two stuffing-boxes.

Taking the pressure on the step A as being the difference between the actual pressure and the pressure on the end

C, we thus eliminate the effect of this latter (end) pressure. We then have the following end pressures :—

End thrust on blades of section	1 =	2,030lbs.
,, ,, ,,	2 =	1,520lbs.
,, ,, ,,	3 =	1,880lbs.
Total end thrust on blades	=	5,430lbs.
Static pressure on step A	=	29,900lbs.
,, ,, ,, B	=	10,000lbs.

The total end thrust to be balanced by the balance pistons is 45,330lbs.

We then make the balance piston D to balance the end thrust on the blades of section 1; the balance piston E to balance the pressure on the blades of section 2, and on the step A; and the balance piston F to

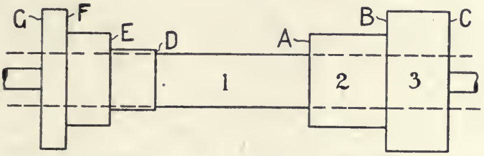


FIG. 206.—SKETCH TO ILLUSTRATE CALCULATION OF BALANCE PISTON AREAS.

balance the pressures on the step B and the blades of section 3. The pressures on these balance pistons are, of course, 178·5lbs., 59lbs., and 10·9lbs. per square inch, so that we have the dimensions as given in Table XLII.

TABLE XLII.

Piston.	Thrust.	Net Pressure.	Area.	Diameters, Inches.	
D	2,030	175·5	11·37	19·6	19·96
E	31,420	59·0	533	19·96	32·8
F	11,880	10·9	1,090	32·8	49·6

The radial clearance over the pistons we may make about ·05in., ·06in., and ·07in. for D, E, and F respectively. The axial clearances in the grooves of the piston we may make 0·02in., 0·015in., and 0·01in., the largest clearance being for the piston D. As to the number of grooves, 25 for D, 20 for E, and 12 for F will be suitable.

The actual clearance when running may be more or less than this, as it is cut down as fine as possible; particularly so when making a test. The leakage over these pistons varies a great deal, although it is believed to be seldom less than 5 per cent. of the total steam consumption except, perhaps, in large turbines. It should be easy to determine it by inserting a plate orifice—large enough not to set up much back pressure, and preferably set in an enlargement of the connecting pipe—in the connecting pipes between the pistons and the steps of the drum, the pressures on the two sides of the orifice being measured.

Blade Pitch.—We shall space the blades so that the axial clearances on the two sides of a row of blades are unequal, as was explained earlier on when dealing with drums.

In Table XLIII. we have the various blading dimensions suitable in this case.

TABLE XLIII.

Section.	Sub-section.	Blade Length.	Blade Width.	Axial Clearances.		Axial Pitch.
1	1	·58	·25	·09	·11	·70
	2	·81	·3	·10	·13	·83
	3	1·02	·3	·11	·15	·86
	4	1·27	·3	·12	·17	·89
2	1	·595	·25	·10	·15	·75
	2	·925	·3	·11	·17	·88
	3	1·405	·5	·15	·22	1·37
	4	1·915	·5	·17	·24	1·41
3	1	1·55	·5	·15	·23	1·38
	2	3·23	·6	·20	·28	1·68
	3	5·41	·7	·25	·34	1·99

The usual practice is to make the axial clearances each equal to half the blade width, thus making the pitch equal to three times the width. The saving due to the adoption of unequal pitches in the above example is only 2·82in., and is probably not worth the extra care in the designing and machining which it entails, although in view

of the fact that the turbine is only using saturated steam it is quite probable that the saving in rotor length could without difficulty be increased to at least 5in. when using superheated steam.

Taking the ratio of the electrical horse-power to the indicated horse-power as $0 \cdot 9$ the maximum indicated horse-power of this turbine is about 5,670. The values calculated from formulæ 1, 2, and 4, as given earlier, are 5,870, 5,470, and 5,230 i.h.p. respectively.

Modification of Calculations.—A few words of criticism as to the above design may be helpful to readers. The above design is not a first-rate one for most purposes, but it will serve to illustrate the method without undue complication and length in the calculations.

It will be noticed that the blade lengths are very short; too short, in fact. This is on account of the blade angles adopted. In the above case, the clearance leakage would be excessive. In order to reduce it, the blade angles should be altered to, say, 70° and 30° (see page 103, Chapter IV.). This would increase the blade lengths by about 135 per cent. The blades in the last two subsections at the low-pressure end would require little, if any, modification in their blade angles. It is common practice to alter the blade angles of the last few rows so as to increase the axial velocity of the steam, and thus do away with the necessity for excessive blade lengths. This is, of course, accompanied by a decrease in the gauging factor (see page 129.)

Under the new conditions as to blade angles, the number of stages will have increased to about 75. It would be quite unusual, however, to use so many under the conditions as to steam pressure and lack of superheat. It is a well-known fact that the number of stages can be appreciably less than the calculated number without any marked loss of efficiency. Such a reduction in the number of stages considerably cheapens the turbine.

Furthermore, the number of stages will depend upon the guarantees as to steam consumption. At light loads the mean effective initial pressure on the turbine side of the stop-valve pressure is always much below the maximum pressure even with a blast governor, and still more so with a throttle governor. Consequently, the number of

stages giving the maximum efficiency at light loads will be considerably less than the number for maximum efficiency at full load. A large number of stages will cause excessive frictional losses at light loads.

Clearly, then, if stringent guarantees are given as to the light-load consumption, the number of stages must be cut down somewhat. Where no special conditions are given the number of stages should be designed for average load conditions. In this case the number of stages would probably be about 70. If steam economy was not of the first importance, somewhat higher axial velocities for the steam could be used and the number of stages appreciably reduced, thus cheapening the turbine.

In our calculations it is, perhaps, desirable to allow a drop of 3 or 4 B.Th.U. at the junction of two sections in order to allow for the losses and the necessary increase in the steam velocity. These can easily be allowed for on the heat diagram. Also, as previously mentioned, the mean effective initial pressure is in general rather less than the maximum stop-valve pressure.

Design for Superheated Steam.—As soon as we know the specific heat of superheated steam and its density at different temperatures, we can complete our heat diagram and proceed with the design exactly as for saturated steam. In the absence of this knowledge we may assume data, such as a constant specific heat of $0 \cdot 6$ and a gaseous law which makes the volume—commencing with the saturated condition—proportional to the absolute temperature.

Another and probably less accurate method would be as follows: Assuming the specific heat to be, say, $\cdot 6$, the extra available work in the steam due to superheating from absolute temperature T_2 to T_1 is

$$0 \cdot 6\ (T_1 - T_2) - 0 \cdot 6\ T_3 \log \frac{T_1}{T_2}$$

where T_3 is the (absolute) temperature of the exhaust and the logarithm is a hyperbolic one. From our blade angles and speeds we calculate the work done per stage and hence the total number of stages in each section. Now obtain the curves of radial distance between rotor and casing as for saturated steam. Reduce the calculated works per stage inversely as the available heats in the

superheated and saturated conditions and design as for saturated steam. For instance, suppose that the available works in the superheated and saturated conditions are respectively 320 and 300 B.Th.U., and that the work per stage in a certain section is 32. Reduce it to 30 and design as for saturated steam.

As regards the calculated radial distances, these should be increased by from 6 to 8 per cent. per 100° Fah. of superheat above those for a turbine passing the *same weight* of saturated steam. The weight actually passing for a given power is, of course, less when using superheated steam.

Impulse Turbines with Losses.—In the reaction turbine the generation of kinetic energy and its partial waste go on side by side, so that the previous method of assuming that some of the kinetic energy is never generated but is converted right away into heat is perfectly justified. In an impulse turbine this is not true, and is only justifiable as a working basis in certain cases. In the impulse turbine we first have a generation of kinetic energy in the fixed blades—nozzles—with only a small loss of from 5 to 10 per cent. taking place concurrently. Then we have the actions in the moving blades with their accompanying waste of kinetic energy, but no generation of energy.

This is represented on the heat diagram (Fig. 202). If the expansion line F S has been obtained as for a reaction turbine it will only give us the steam conditions at the *entrances* to the fixed blades, just previous to the commencement of a new stage. For many purposes this is all we want to know. Suppose, however, we want to represent the actions and losses in the individual stage, and suppose that the hydraulic loss is 30 per cent. Let us neglect radiation, and assume that the losses are distributed as follows :—

	Per cent.
Loss in nozzles	8
Nozzles to wheel	5
In wheel	10
Wheel to succeeding nozzles	7
Total	30

Draw N Q equal to the losses after leaving the nozzles, that is, make N Q to P Q as 22 is to 70. Then N P represents the kinetic energy leaving the nozzles, and from it and knowing the inlet velocity, we can determine the outlet velocity by projecting on to a velocity curve in the manner already described. Take points R and L in N P such that N R, R L, and L Q represent the 5, 10, and 7 per cent. losses previously mentioned. Then R gives us the condition of the steam as regards velocity, volume,

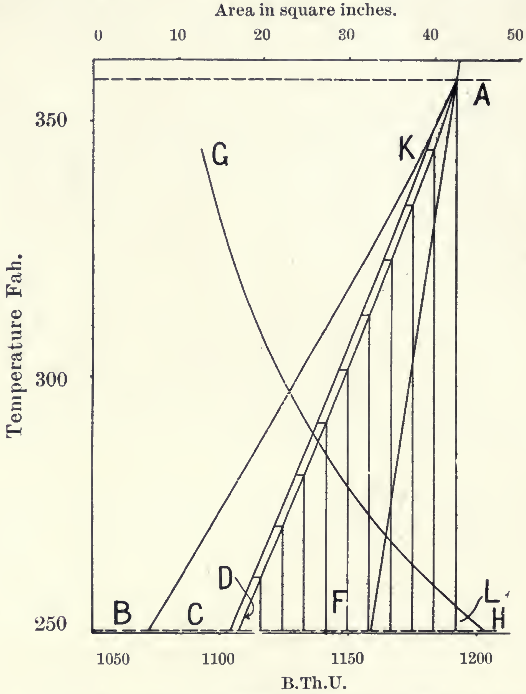


FIG. 207.—HEAT DIAGRAM FOR DESIGN OF IMPULSE TURBINE.

and temperature, entering the moving blades, L the condition on leaving the moving blades, and Q the condition on entering the next set of nozzles. We have assumed that the pressure in the moving blades is constant which cannot be quite true with the ordinary blade formations. The necessary modification to the above in order to allow for this fact will suggest itself to the reader.

If we determine the curve on which the points N for the various stages lie, we can use it in calculating the outlet areas of the fixed blade openings. As these areas are the most important ones in an impulse turbine, it is, perhaps, advisable to do so. In this case it must be remembered that the axial steam velocity is greater than that given by the ordinary velocity diagram unless allowances have been made on it for point-to-point losses.

A similar analysis of the losses in each stage of a Curtis turbine can be performed. In either case the method of calculating the number of stages is the same as that adopted for the reaction turbine.

Design for Impulse Turbine.—In order to illustrate a little more definitely the design of an impulse turbine we will work out roughly the figures for the high-pressure cylinder of a 1,000 kw. turbine capable of developing a maximum load—without by-pass—of 1,200 kw. at 24lbs. per kw. hour. We have the following data :—

Maximum load = 1,200 kw.
R.P.M. = 1,500.
Stop-valve pressure = 150lbs. absolute.
Exhaust pressure = 30lbs. absolute.
Gross available B.Th.U. per lb. of steam = 119·6.
Radiation and final exhaust loss = 5·6 B.Th.U.
Net available B.Th.U. = 114.
Hydraulic efficiency = 65 per cent.
Indicated work = 74 B.Th.U.
Work per stage = 7·4 B.Th.U.
Stages = 10.

Assume the following approximate losses per stage :—

	Per cent.	B.Th.U.
Loss in nozzles	6	0·69
Nozzles to wheel	10	1·14
In wheel	11	1·25
Wheel to nozzles	8	0·92
Total	35	4·0

Taking the blade angles to be as follows :—

Inlet to nozzles = 85°
Outlet from nozzles = 25°
Wheel, both = 40°

The peripheral speed of the moving blades must, with these angles and losses, be 285ft. per second, in order to give an indicated work per stage of 7·4 B.Th.U. We then have the following steam velocities :—

Inlet to nozzles	= 163ft.	per sec.
Outlet from nozzles	= 750ft.	,,
Inlet to wheel, absolute	= 715ft.	,,
,, ,, relative	= 480ft.	,,
Outlet from wheel, absolute	= 270ft.	,,
,, ,, relative	= 415ft.	,,

The inlet depth of the ports in the diaphragms should be somewhat greater than the outlet depth, or the inlet edges should be well rounded, as the above ratio of outlet to inlet velocities for the nozzles is too great to be accommodated if the radial depth is the same at inlet and outlet, unless the nozzles have the effect of a sharp inlet.

Fig. 207 is the heat diagram for this turbine. *A B* is the adiabatic line from dry steam at 150lbs. absolute. *A L* is a (vertical) throttling line, and *A D* is the expansion line showing the condition of the steam at inlet to the several sets of nozzles. This line is obtained in the manner previously described for a reaction turbine—that is, by making its distance from the throttling line *A L* equal to 65 per cent. of the net available work, plus the radiation loss, which in this case we take to increase fairly uniformly with a drop in temperature, up to a maximum of about 5 B.Th.U. at exhaust. The losses in each stage between the outlet of one set of nozzles and the inlet to the next set amount to 3·3 B.Th.U., so that by drawing the line *K C* distant 3·3 units from *A D* we obtain the expansion line, showing the condition and volume of the steam at the outlets from the nozzles. In this case the distance between the lines *A D* and *K C* is small—hardly worth allowing for, in fact—but would be much greater in a Curtis turbine.

Divide the base line *D L* into 10 equal parts, and project vertically on to the line *A D*. From these latter intersections project horizontally on to *K C*, the points thus obtained giving us the steam condition at the outlets from the nozzles. From the steam volumes thus obtained we calculate the outlet nozzle areas. These areas are given

in Table XLIV., and it should be noted that they are measured in planes perpendicular to the shaft.

The axial velocity of the steam at outlet is 750 sin 25° or 319ft. per second, and the weight of steam passing is 8lbs. per second. The areas are given in square inches. The method of making the calculation is obvious from the Table :—

TABLE XLIV.

No.	Pressure, Absolute.	Temp., Fahr.	Heat units requir'd to dry steam.	Cubic feet per B.Th.U.	Volume less than dry steam.	Volume of dry steam.	Actual volume, cubic feet.	Outlet area in square inches.
	p	t	H	Q	Q H	U	u	A
1	125·7	344·5	7·2	·00400	·03	3·50	3·47	12·5
2	108·1	333·7	12·4	·00458	·06	4·04	3·98	14·4
3	93·9	323·0	17·7	·00518	·09	4·60	4·51	16·2
4	80·4	312·2	23·0	·00595	·14	5·34	5·20	18·8
5	68·7	301·5	28·1	·00684	·19	6·18	5·99	21·6
6	58·7	291·1	33·5	·00788	·26	7·18	6·92	25·0
7	50·1	281·0	38·5	·00906	·35	8·33	7·98	28·8
8	42·6	270·9	44·0	·01047	·46	9·70	9·24	33·3
9	36·0	260·9	49·3	·01218	·60	11·36	10·76	38·8
10	30·0	250·3	55·0	·01435	·79	13·48	12·69	45·8

The curve *G H* in Fig. 207 gives us the nozzle (outlet) areas when measured in planes perpendicular to the shaft.

From the calculated peripheral speed and the revs. per minute we see that the mean diameter of the blade circles—all same diameter—is 3ft. 7·5in.

Entropy Diagrams.—As many engineers are not at all clear as to what an entropy diagram is, a little space will be devoted to explaining this very useful tool of the steam engineer.

There is no need to worry as to what entropy is. For our purposes it is merely a mathematical quantity, a relation between quantities of heat and degrees of temperature. If we add heat to a body we increase its entropy and conversely, if we abstract heat we reduce the entropy. This change in entropy is in exact inverse ratio to the absolute temperature at which the change takes place and

is proportional to the quantity of heat added or abstracted. It is convenient to take the entropy of a body at 32° Fah. (0° C.) as zero.

Then, if
T is the absolute temperature.
δT the increase in temperature.
δH the increase in heat.
$\delta \phi$ the increase in entropy.
k the specific heat.
L the latent heat.

$$\delta \phi = \frac{\delta H}{T}.$$

This simple equation holds good under any and all circumstances. Except in such cases as the evaporation of water, where the temperature remains constant whilst the heat is being added, the addition of heat to a body raises its temperature, but by taking the increase in heat (δ H) very small the increase in temperature which it produces will be so small that the absolute temperature T is not appreciably altered. Now we know that

$$\delta H = k \delta T$$

hence

$$\delta \phi = k \frac{\delta T}{T}$$

and

$$\phi = k \log T,$$

the logarithm being, like all others used in entropy equations, a hyperbolic logarithm.

Then, if the temperature rises from T_2 to T_1 the increase in entropy will be

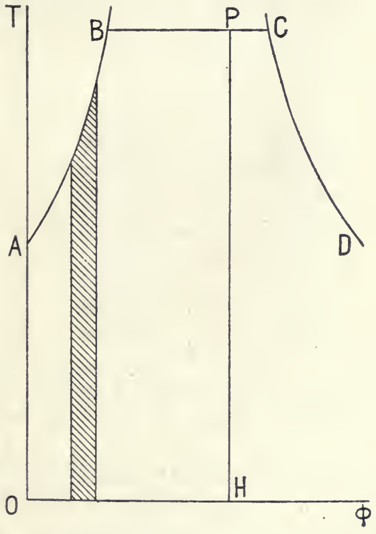
$$\phi_1 - \phi_2 = k \log \frac{T_1}{T_2}.$$

When k is variable a somewhat more complicated formula results. The increase in entropy due to the evaporation of unit weight of water will evidently be

$$\frac{L}{T}.$$

A table of entropy values is given on page 364. To construct an entropy diagram we proceed as follows: Take rectangular axes O T and O ϕ, Fig. 208, temperatures being measured along the former, entropy values along

the latter. From our tables we know that at a certain temperature the entropy of one pound of water is ϕ_w. In this way we draw in the "water line" A B. Similarly our tables give us the values of the total entropy for steam, from which we can draw in the "steam line" C D. As in the heat diagram previously described, all points to the right of the steam line represent superheated steam, and all points between the steam and water lines represent



Absolute Zero.

FIG. 208.—ENTROPY DIAGRAM FOR STEAM.

mixtures of steam and water. Thus, if in Fig. 208 P represents the condition of the steam, then evidently

$$\frac{PC}{PB} = \frac{\text{water}}{\text{steam}},$$

and

$$\frac{BP}{BC} = \text{dryness fraction.}$$

It will be clear that we can easily draw on the diagrams constant-dryness, constant-pressure, constant-volume, and constant-entropy lines. The latter will be vertical and will, moreover, be adiabatic lines. Since the total heat of the steam at any point is known, we can also

draw in constant total-heat (throttling) lines; or, as we might perhaps call them, isothermic lines. These will be found very useful in steam turbine problems.

Table of Entropy for Steam.

t = Temperature Fahrenheit.
ϕ_w = Entropy of one pound of water from 32 deg.
ϕ_s = Entropy of formation of steam $= \frac{L}{T}$.
$\phi = \phi_w + \phi_s$ = Total entropy of one pound of dry steam.

t	ϕ_w	ϕ_s	ϕ
32	—	2·2189	2·2189
60	0·0553	2·0621	2·1174
100	0·1296	1·8649	1·9945
150	0·2154	1·6547	1·8701
200	0·2949	1·4760	1·7709
250	0·3690	1·3220	1·6910
300	0·4385	1·1880	1·6265
350	0·5042	1·0698	1·5740
400	0·5665	0·9649	1·5314

NOTE.—L = Latent heat. T = Absolute temperature in Fahrenheit deg. $= 461 + t$.

Now, from our first entropy equation we see that

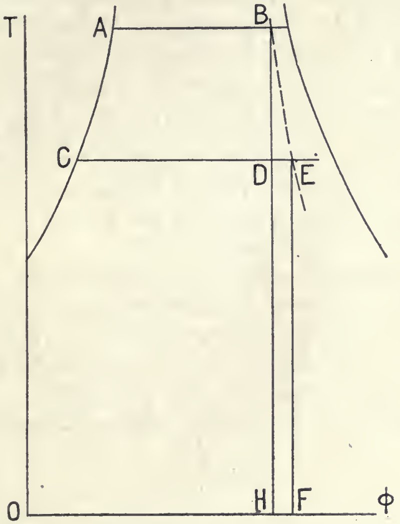
$$\delta H = T \delta \phi$$

That is, the increase in the heat is represented by the area under the curve and bounded by the two entropy ordinates, as is illustrated by the shaded area in Fig. 208.

The whole area O A B P H evidently represents the total heat added from 32° Fah. up to the point P. In practice it is convenient to cut off all the diagram below say 60° or 100° Fah., but in doing so we must not forget the cut-away portion or we shall be liable to get our expansion lines entirely out of shape.

The construction of an expansion line may be indicated. Suppose that at each point the loss of kinetic energy is, say, 30 per cent. of the theoretically available energy. Then referring to Fig. 209 the pressure (or temperature) falls from A B to C D. The theoretically available kinetic

energy generated by this expansion is represented by the area A B D C; 30 per cent. of this is converted into heat, hence we must find a point E such that the area D E F H is 30 per cent. of the area A B D C, D E F H representing



Absolute Zero.

Fig. 209.—Construction of (Turbine) Expansion Line.

this heat. Then E is on the expansion line through B. The area of the entropy diagram for a reciprocating engine is equal to its indicated work, but this is not true of the diagram for a turbine, because the less work done the more heat produced (by loss of kinetic energy) and hence the larger the diagram.

Entropy Diagram for Superheated Steam.—At present we are hopelessly in a fog as regards the properties of superheated steam. In particular we want to know the specific heats at constant pressure and constant volume. In what follows we shall refer to the specific heat at constant pressure merely as the specific heat.

A great many attempts have been made to determine this specific heat. Regnault obtained a mean value between 437° and 257° Fah. of 0·48, and this is very generally accepted in practice, although it is most probably not correct. Between 256° (Fah.) and 212° Mr. Macfarlane

Gray gives the value 0·378. Prof. Carpenter in 1904 obtained (experimentally) the formula for the specific heat—

$$K = 0 \cdot 46 + 0 \cdot 0015\, p.$$

p being the absolute pressure ; no appreciable variation due to temperature being observed. In 1904 Greissmann obtained the formula

$$K = 0 \cdot 00222\, t - 0 \cdot 116$$

the specific heat being thus, according to him, independent of the pressure ; t is the temperature Fahrenheit. In 1899 Grindley obtained values of the specific heat which increased with the temperature but showed no appreciable variation with the pressure.

Prof. Callendar in 1900 found that the specific heat increased with the pressure but decreased with an increase in the amount of superheat. Thus in Table XLV. we have some of his figures for the specific heats at the saturation points, and also for the specific heat at constant atmospheric pressure but different temperatures.

TABLE XLV.

Temperature Fah...	32	104	140	212	248	284	320	392
At saturation... ...	·497	·501	·505	·524	·539	·558	·582	·640
Atmospheric pressure	—	—	—	·524	·518	·514	·510	·506

About two years ago Prof. Lorenz obtained results of a similar character, and gave the following formula :—

$$K = 0·43 + 1{,}480{,}000 \frac{p}{T^3}.$$

More accurate experiments are necessary before the results can be finally accepted.

In 1903-4 the American General Electric Company experimented upon a full-sized steam turbine, accurately measuring the brake horse-power, the radiation, and leakage, the input to the turbine, and the output to the condenser. The initial steam pressure was 155lbs. per square inch absolute, and the average values of the specific heat from saturation temperature to the temperature in the table are given below. (Table XLVI.).

TABLE XLVI.

Superheat, Fah. ...	0	100	150	200	250
Specific heat	0·52	0·65	0·7	0·74	0·77

Prof. Denton, discussing some recent experiments at the Munich Polytechnic, points out that the specific heat of superheated steam—according to these experiments—increases with the pressure and decreases as the amount of superheat increases. Some of Prof. Denton's results are given in Table XLVII.

TABLE XLVII.

Pressure, Pounds per Square Inch Absolute.	Temperature, Fah.	Mean Specific Heat at Constant Pressure. Range of Superheat.		
		18° Fah.	90° Fah.	180° Fah.
99·48	327	0·567	0·551	0·537
139·32	352	0·597	0·577	0·559
190·70	378	0·634	0·609	0·586
266·20	406	0·686	0·656	0·626

In view of the contradictory results so far obtained, it is perhaps as well to take a constant value for the specific heat under all conditions, say, 0·55 or 0·6. It is to be hoped that the experiments now being conducted at the National Physical Laboratory will give some reliable results. So far the weight of evidence seems to favour the type of results obtained at Munich as outlined above.

We also want to know the specific heat at constant volume, or data respecting the densities of superheated steam. The simplest rule is to calculate the volumes on the assumption that the product of the pressure and volume is proportional to the absolute temperature. This rule is probably considerably in error for low values of superheat. Prof. Perry gives the formula for steam in any condition :—

$$u = \frac{85{\cdot}5\ T}{p} - \left(0{\cdot}118 + \frac{3{,}200}{p + 3{,}350}\right)$$

where T = absolute temperature in Fahrenheit degrees.

p = absolute pressure in pounds per square foot.

u = volume per pound in cubic feet.

Prof. Denton gives the following formula :—

$$p\,u = B\,T - p\,(1 + a\,p)\left[\,C\left(\frac{373}{T}\right)^3 - D\right]$$

where p = kilogrammes per square metre.
u = volume of kilogramme in cubic metres.
T = absolute temperature, Centigrade.
B = 47·1.
a = 0·000002.
C = 0·031.
D = 0·0052.

Entropy Diagram for Gases.—Our knowledge of the properties of gases at high pressures and temperatures is too uncertain to enable us to be very dogmatic, so that it is perhaps best to assume that they act like perfect gases and follow the law

$$p\,u = R\,T$$

Then if
k_p = specific heat at constant pressure
k_v = specific heat at constant volume
the increase of entropy at constant pressure will be

$$\phi - \phi_0 = k_v \log \frac{T}{T_0} + R \log \frac{u}{u_0}$$

and at constant volume it will be—

$$\phi - \phi_0 = k_p \log \frac{T}{T_0} - R \log \frac{p}{p_0}$$

For 1lb. of air R has the value 53·18, T being the absolute temperature in Fahrenheit degrees and p the pressure in pounds per square foot; also k_p = 0·238 and k_v = 0·169.

To find the specific heat of a mixture of gases we multiply the weight of each gas by the specific heat of that portion; add these products together and divide the sum thus obtained by the total weight of the gases. This merely expresses the fact that the heat required to raise the temperature of the mixture a certain amount is equal to the sum of the heats required to raise the separate portions the same amount.

Constant Heat Lines.—On any entropy diagram we can draw in lines connecting up all points having the same total heat. These lines are throttling lines, and are useful in many ways, especially when combined with

velocity curves after the manner described in connection with the heat diagram. If, for instance, the expansion curve cuts—in order—constant heat lines H_1 and H_2 we know that the amount of heat converted into work, kinetic energy, or lost by radiation is $H_1 - H_2$. This applies equally to a reciprocating engine or a turbine.

Work Diagram for Gases.* —The exact construction of the diagram depends upon the nature of the problems to be solved. As we are only here concerned with gas-turbine and turbo-compressor calculations, we shall only describe the applications of the diagram or diagrams to such cases.

There are really two diagrams, of which one is more particularly useful for turbine work, although it can

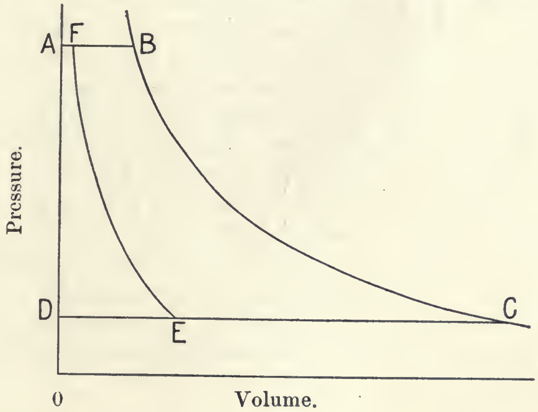


FIG. 210.

easily be made to serve for all reciprocating engine problems. By means of the diagram we can at once read off the pressures, volumes, and temperatures at all points during the cycle, and the diagram gives us directly the efficiency, the work done, the heat received, and other similar quantities.

In Fig. 210 we have an ordinary pressure-volume diagram, in which B C is a portion of the expansion—or compression—curve. For a turbine (or turbo-compressor) working between the pressures O A and O D, the area A B C D represents the theoretical work done over the

* For further details see "A New Work Diagram for Gases," by Frank Foster, "The Engineer," Dec. 1st, 1905.

complete cycle. It also represents the work done in a reciprocating compressor or hot-air engine without any cylinder clearance and with external compression to the pressure O A. Consider only turbine problems for the time being, and suppose that the expansion curve to be of the form

$$p\,u^n = \text{a constant},$$
or
$$p^m u = C$$

where m is the reciprocal of n.

Further, this expansion curve is the theoretical, *not the actual*, expansion (or compression) curve. In general then this curve represents adiabatic expansion. If we take into account the effect of radiation or a jacket the curve will not be an adiabatic line. Still, it is quite easy to determine its equation provided only we know the loss to, or gain of heat from external sources. Thus, if we consider one pound of air, and we know that during expansion to a certain pressure the water jacket has removed 23·8 B.Th.U., we see that the final temperature is 100° Fah. below that corresponding to adiabatic expansion. Therefore its volume is reduced by an amount which is easily calculable from the ordinary gaseous law. In this way we can quickly determine the value of the index m. The actual curve in a turbine will not in any case actually have this theoretical expansion curve owing to the conversion of kinetic energy into heat, which will in all cases increase the volume above that calculated

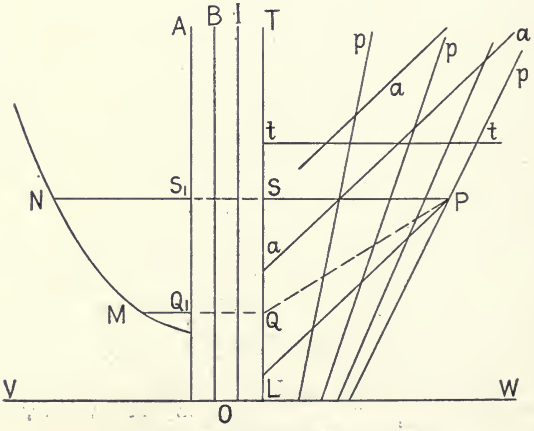


FIG. 211.—WORK DIAGRAM FOR GASES.

by the theoretical expansion formula, whether adiabatic or not.

In determining the equation to this theoretical-expansion curve we must make allowances for all heat given to or taken from the turbine by external sources, but not for any internal friction and eddy losses with their resultant heating effect.

Then if

p =lbs. per square foot absolute.
u =volume in cubic feet per pound.
T =absolute temperature on Fahrenheit scale.
=461 + temp. Fah.
R =a constant $= K_p - K_v$.
k_p =specific heat at constant pressure in B.Th.U.
k_v = ,, ,, ,, volume ,,
K_p = ,, ,, ,, pressure in ft.-lbs.
K_v = ,, ,, ,, volume ,, ,,

We have $p\,u = R\,T$.

The values of the various constants for different gases are given in the appendix. (Table LIX.)

We know also that the area A B C D (Fig. 210)

$$= W = \frac{p_1 u_1 - p_2 u_2}{1 - m} \quad \dots\dots\dots \quad (1)$$

$$= \frac{C\,(p_1^{1-m} - p_2^{1-m})}{1 - m} \quad \dots\dots \quad (2)$$

$$= \frac{R\,(T_1 - T_2)}{1 - m} \quad \dots\dots\dots \quad (3)$$

Also $$C = p^m u = \frac{p\,u}{p^{1-m}}$$

$$= \frac{R\,T}{p^{1-m}} \quad \dots\dots\dots\dots \quad (4)$$

Clearly, then, the constant-pressure line showing the relation between W and T for any given pressure is a straight line through the origin of absolute temperature. To determine its inclination we determine the value of C at any known pressure and temperature from equation (4). Inserting the value thus found, in equation (2) we are enabled to draw in the constant-pressure line. The constant-pressure lines on our work diagram (Fig. 211) have been thus obtained. The pressures marked on them are, however, in terms of pounds per square inch, whereas pounds per square foot must be used in the calculations.

We have, of course, to take some standard back-pressure p_2, usually atmospheric (14·7lbs. per square inch or 2,116lbs. per square foot). The constant-pressure line p_2 will, of course, coincide with the axis of temperature.

Next we can draw in the constant-volume lines marked u on the diagram. The temperature at which a constant-volume line crosses a constant-pressure line is, of course, readily calculated from the formula $p\,u = R\,T$. Again, if a quantity of heat H (or its work equivalent) is added at constant volume, the rise of temperature is $\frac{H}{k_v}$ (or $\frac{H}{K_v}$ if in work units). Similarly, the temperature rise at constant pressure would be $\frac{H}{k_p}$. We can then put in two scales B and A to represent respectively the quantities of heat—or their work equivalent, or both—corresponding to the temperatures for constant-volume and constant-pressure conditions during the addition of the heat.

Lines of constant $p\,u$ are constant-temperature or isothermal lines. We can therefore add a scale—I on the diagram—showing these products.

Again, equation (3) shows us that the expansion line on the work diagram will be straight with a constant inclination for any one gas and index m. Hence the expansion lines on our diagram (marked a) will be parallel straight lines. Suppose that a fraction x of the energy of expansion represented by the area A B C D to be converted into heat instead of useful kinetic energy. The temperature will thereby be raised from T_2 to T_3 and we have

$$K_p\,(T_3 - T_2) = x\,W = \frac{x\,R\,(T_1 - T_2)}{1-m} = x\,K_p\,(T_1 - T_2)$$

This shows that if x is the same at all points during the expansion the true expansion line will also be straight.

In the particular case where the gas is merely throttled $x = 1$, and we obtain a series of throttling lines (t) which are horizontal straight lines of constant temperature.

Now we know that the change in kinetic energy for a loss x W during the free expansion is

$$\frac{v_2^2 - v_1^2}{2\,g} = W\,(1-x) = \frac{R\,(T_1 - T_2)\,(1-x)}{1-m}.$$

We can construct a series of curves, all parabolas from the same template, showing the relation between the

velocity and the work of expansion. This can perhaps be best illustrated by an example. In Fig. 211 we start from P. P L is the expansion line when no loss of kinetic energy occurs ; P Q the actual expansion line, and P S the throttling line. Then, measured on the constant-pressure scale A, S_1 Q_1 is the actual work converted into kinetic energy. M N is a velocity curve showing the relation between the velocity (scale O V) and the work equivalent of it O A. Project Q and S on to O A and make Q_1 M equal to the initial velocity (v_1). Draw a velocity curve through M to meet S N in N. Then S_1 N is the final velocity (v_2). We will now illustrate the use of the diagram.

These velocity curves are merely parabolas showing the relation between a given quantity of heat and the velocity change caused by it. They are all alike, and quite independent of any losses in the working fluid. Table LXIV. in the Appendix gives all necessary data for drawing these curves. In the particular example given above, it would have been better had the velocity curve been reversed, Q_1 M replacing S_1 N, so that the velocities correspond directly with the temperatures at which they occur. Thus, in the above example, the diagram gives the initial velocity at the final temperature.

Gas Turbine or Turbo Air Compressor of the Rateau Type.— For the moment we will consider only the expansions and neglect the other operations in the cycle. In our figure Q P is again the actual expansion line. First consider a turbine. The gas expands and the "indicated" work will be S Q. Commence at S and mark off along S Q— along S_1 Q_1 really, but as they have to be projected on to S Q it is simpler to speak of them as along S Q—quantities equal to the indicated works done in each stage. Where the diameter of the wheels is constant these quantities will be equal. Draw lines through these positions marked off along S Q, parallel to the throttling line S P and intersecting the expansion line P Q. The points of intersection give us the pressures, volumes, and temperatures in the respective stages. The velocities generated in each stage are obtained from the velocity curves by the method previously described. For a turbo-compressor a slightly different construction is necessary (Fig. 212). Neglecting the kinetic energy losses we obtain the compression line

L S as for a turbine. But the amount of energy x W converted into heat raises the temperature from L to P just as in expansion the temperature is raised from L to Q. Draw the throttling line P Q, and on the scale A the amount of "indicated" work done in the compression

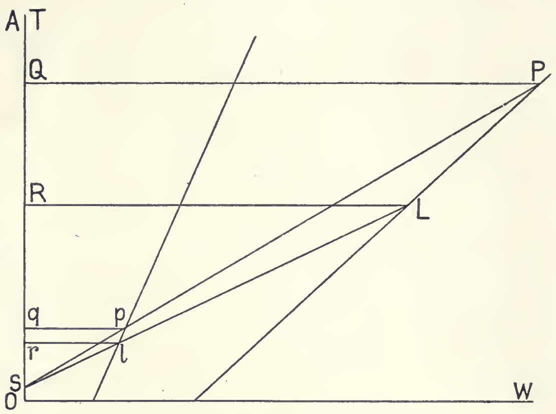


FIG. 212.—TURBO COMPRESSOR DIAGRAM.

is S Q. The work theoretically required is S R, obtained by drawing the throttling line L R. This diagram (Fig. 212) brings out a result of the greatest importance in turbo-compressor work. Consider two compressors compressing to the pressures P L and $p\,l$ respectively. The efficiencies of compression are respectively

$$\frac{R\,S}{Q\,S} \text{ and } \frac{r\,S}{S\,q}.$$

Now, since the pressure line P L has a greater inclination to the vertical than the pressure line $p\,l$, a little consideration of the geometry of the diagram will show that *the efficiency of the high compression is less than that of the low compression ;* a most important result in turbo-compressor practice, as it indicates that at high pressures *at least*, the turbo-compressor will hardly be able to compete with the reciprocating compressor in point of efficiency. Probably, too, the compression line S P will be steeper the higher the pressure attained.

Parsons Type.—The expansion or compression here takes place partly in the fixed and partly in the moving

blades, usually half in each. The method of constructing the diagram is precisely that previously described, except that we now have virtually twice as many stages as there are sets of moving blades.

Rateau and Curtis Types with Losses.—In our previous consideration of the Rateau and Curtis turbines we had grouped all the losses in each stage together, whereas they take place continually as the gas passes from the entrance of the nozzles in one stage to the entrance of the nozzles in the next stage. In what follows we shall assume that the losses take place at constant pressure, which is, however, probably not true.

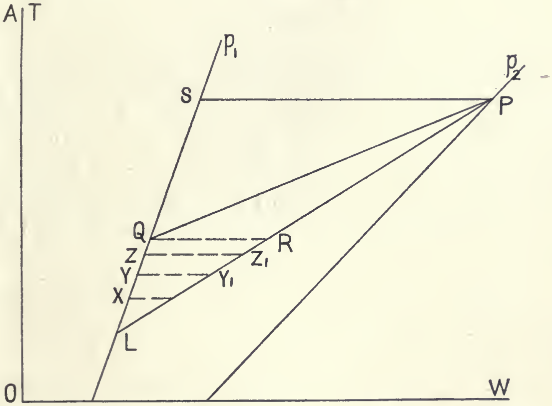


FIG. 213.—IMPULSE GAS TURBINE WITH LOSSES.

If we do not like this assumption we may modify the diagram (Fig. 213) by drawing the throttling line Q R and projecting X Y and Z on to Q R. Neglecting this for the time being, first obtain the points on the expansion line corresponding to the inlet and outlet of each stage in the manner previously described. Now consider one of the stages (Fig. 213). The pressure falls from p_1 to p_2. P S is the throttling line, P L the theoretical expansion line—usually adiabatic—and Q the final condition of the gas obtained from the actual expansion line—of which P Q is a portion—for the turbine as a whole. Suppose that we are dealing with a Rateau turbine: the loss, which

is proportional to Q L, being, say, 38 per cent. of the available work and distributed as follows :—

	Per cent.
In nozzle	8
Between nozzle and wheel	12
In wheel	11
Between wheel and nozzle	7
Total in stage	38

Then mark off L X, X Y, &c., proportional to these losses, in order (nozzle loss, nozzle-wheel loss, &c.). Then the points P, X, Y, Z and Q respectively give us the conditions at entrance to the nozzle, exit from the nozzle, entrance to the wheel, and so on ; Q being the condition at entrance to the next set of nozzles. The same method of procedure must be adopted for a Curtis turbine, the only difference being that there are now several wheels and guides so that we have a correspondingly greater sub-division of the losses.

The above remarks apply to turbines. The necessary modifications of the diagrams in order to make them apply to turbo-compressors have already been indicated.

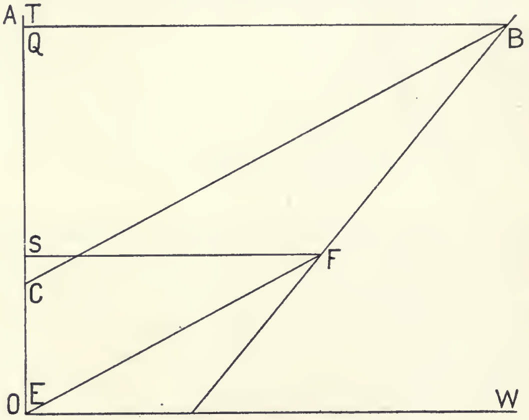


FIG. 214.—GAS TURBINE ; WORK DIAGRAM.

We will illustrate the use of the work diagram by considering the case of a gas turbine and compressor combined. Fig. 210 is the pressure-volume diagram. The compressor compresses the gas from E to F. Heat

is then added—either by combustion or from an external source—at constant pressure, raising the temperature and increasing the volume to B, when expansion takes place to C. Fig. 214 is the work diagram for this cycle; E F is the true compression line; a quantity of heat H—in work units on the scale A—is then added, raising the temperature to B, when expansion takes place along B C. B Q and F S are throttling lines. Then we have the following:—

Heat received	$= H = Q\,S$
Work done on gas	$= S\,E$
Work done by gas	$= Q\,C$
Net work done	$= Q\,C - S\,E$
Thermal efficiency	$= \dfrac{Q\,C - S\,E}{Q\,S}$

It may be as well to remind the reader that the expansion curves B C and F E in the pressure-volume diagram Fig. 210 are the theoretical curves from which W has been calculated. In the case of a reciprocating compressor E F is also the actual compression line and the area A F E D represents the work spent in compression.

In Fig. 215 we have a scale work diagram for air. The specific heats have been assumed constant at all temperatures, although it is probable that they increase with the temperature. The back pressure is atmospheric.

In order to assist those who desire to construct a work diagram Table XLVIII. has been compiled. It shows the values of W in foot-pounds per pound of air when the temperature of the air is 2,000 absolute degrees

TABLE XLVIII.

Lbs. per square inch. p	Work of Expansion. Ft.-lbs. W	p	W	p	W
—	—	25	52,400	80	142,500
—	—	30	68,900	90	150,000
14·7	—	35	81,600	100	156,100
16	9,000	40	92,600	150	181,000
17	15,000	45	101,500	200	195,000
18	21,000	50	109,200	250	206,000
19	26,000	60	122,500	300	214,000
20	30,850	70	133,600	350	220,000

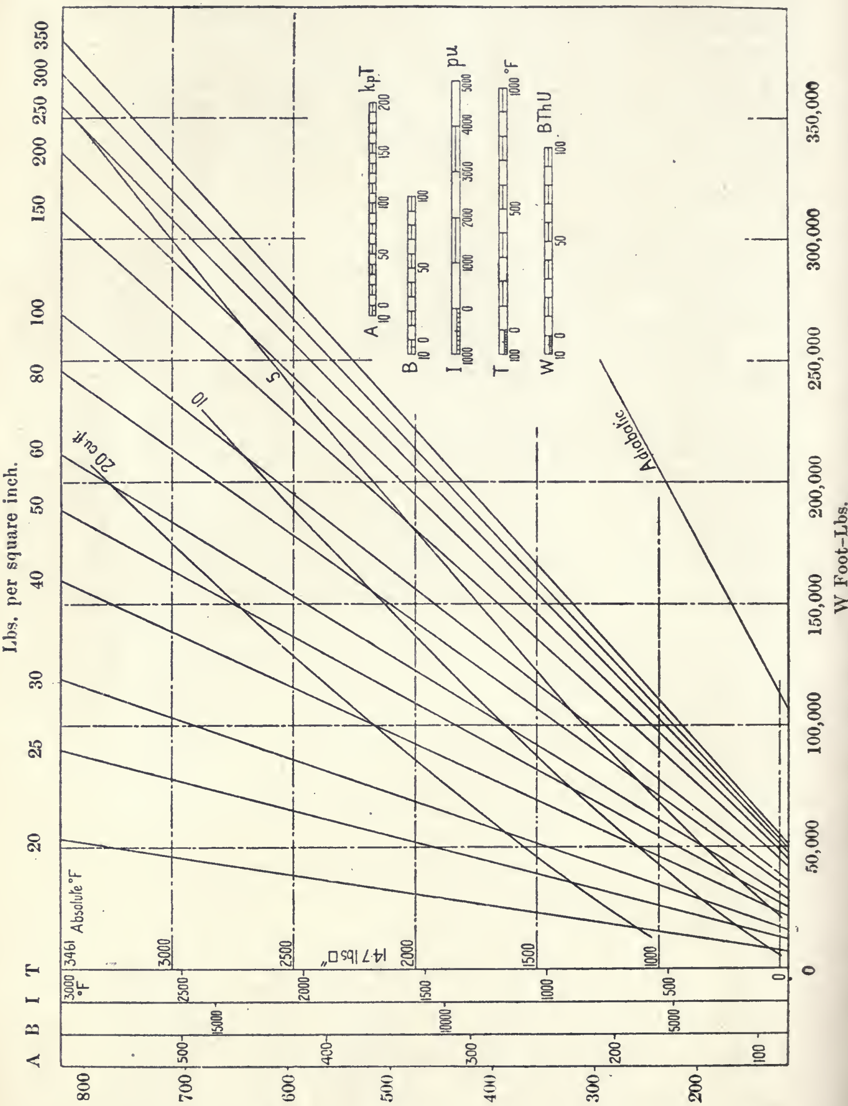


Fig. 215.—Work Diagram for Air.

on the Fahrenheit scale or 1,539° Fah. Thus in order to draw in the constant-pressure line for 200lbs. per square inch we see from our table that *W* is 195,000. Measure off 195,000 ft.-lbs. horizontally at the temperature of 2,000° Fah. absolute and draw in the straight line through this point and the zero of absolute temperature. This is the line required.

Chapter XIV.

CONSTRUCTION AND MISCELLANEOUS.

General Principles of Design.—The distinction between a turbine and other steam machinery which requires to be kept in mind all through the design is that in the turbine the most important members move relatively to one another, but do not touch. At the same time the clearance between these members has in general to be reduced to the smallest possible dimensions.

In order that this may be accomplished with safety it is imperative that as far as possible distortion by the heat should be avoided, and where it cannot be avoided it must be allowed for. Thus it is very necessary to design a casing, or carcass, as it is frequently called, which will not hog or sag, or lose its circularity of cross-section. To a certain extent this can be accomplished by thoroughly lagging the casing and by putting in plenty of metal. The latter method increases the cost and weight of the turbine, and is at best only a partial defence against the enemy of unequal expansion. The turbine casing is split horizontally, and the flanges which are therefore necessary pull the casing out of shape when under steam. In order to counterbalance the action of these flanges, corresponding ribs are sometimes placed along the top and bottom of the cylinder. So far as possible, the casing or cylinder should be quite symmetrical. If stiffening ribs are added to the lower half of the casing, they should be added to the upper; and when such ribs are added, it is much better to employ several equidistant from each other rather than only one or two.

It is extremely desirable that careful micrometer measurements should be taken on actual casings when running under steam, so as to obtain assistance in the design of future casings, and in order to allow of any possible adjustments being made in order to allow for the distortion. As far as possible all turbines should be

run under steam before shipment, and the clearances finally adjusted, so that there is no contact at any point.

When a new design of cylinder is being tried it may

FIG. 216.—BOTTOM PORTION OF CASING OF BRUSH TURBINE.

be advisable to cast a number of extra ribs on the cylinder which are not necessary to its strength and rigidity. When under steam on a testing bed, micrometer readings

should then be taken to determine the distortion. Portions of these extra ribs can then be cut away until the distortion is practically eliminated.

In order to resist any deviation from the circular cross-section the mean temperature from inside to outside of the metal should be as nearly as possible the same at all points round the circumference of any (imaginary) transverse section; and where this cannot be attained, the variations of this mean temperature should occur at regular and fairly frequent intervals round the circumference. Thus, eight equally-spaced fluctuations in temperature between 150° and 90° will give a better result than two of the same magnitude.

The horizontal longitudinal joint is a source of considerable distortion, because in addition to the distortion it effects in the temperature distribution, there is also a break in the continuity of the metal. An attempt is sometimes made to remedy this last defect by staggering the bolt holes at the joint, thus providing virtually two rows of bolts with which to resist any tendency to gape at the joint.

As an instance of the use of stiffening ribs for a large casing, the illustration of the low-pressure turbine casing of the "Carmania" previously given should be referred to.

In order to resist hogging or sagging, short stiffening ribs are usually added at the steps in the diameter and at all circumferential joints. The steam chest between the stop valve and the first row of blades should be symmetrically disposed relative to the shaft. All pipes or connections to the casing have to be carefully arranged, or they will pull the casing out of shape. For instance, there are three pipes on the under side of most reaction turbines connecting the low-pressure and intermediate balance pistons to their respective sections of the blading. The pipe to the intermediate piston is usually incorporated with the casing and forms a stiffening rib, although, owing to its temperature being lower than the mean casing temperature between the two points connected, it tends to make the casing hog. It would probably be advantageous to have a separate connection. The connecting pipe to the low-pressure piston is almost always separated from the casing except at the ends, and

owing to its low temperature an expansion piece is usually inserted.

Most turbines hog. This is partly due to the positions of the feet carrying the cylinder; but it is also largely due to the design of the cylinder body, in particular to the placing of the steam connecting pipes previously mentioned, on the lower half of the cylinder. Placing the admission valves and other gear on the top of the cylinder —as in many of the Parsons and Westinghouse turbines— does a good deal to counteract this hogging by cooling, and therefore contracting the top side of the cylinder. Perhaps the most effective method would be to cast one or more of the connecting pipes in the top half of the cylinder body.

As a rule there are really two connecting pipes to the last step on the blading, arranged one on either side of the vertical centre line. In addition there is a pipe connecting the main exhaust space to the exhaust side of the balance pistons. It has an expansion piece fitted. Owing to these relatively cool connecting pipes and the cooling effect of the bedplate, most turbine casings hog, that is, the centre rises relatively to the ends. In order to remedy this to some extent, the connecting pipes should be separated from the casing, or some of them should be fitted to the top half of the casing, and the connections to the bedplate should be at the coolest points on the casing. Considerable care is necessary in arranging for piping connections to the cylinder, and the main steam piping from the boilers must be carefully designed or it will distort the turbine casing.

After the inside skin has been turned off the casing, it should be closed up and soaked in steam (preferably superheated) before the final boring and planing of the joints, as otherwise it will undergo a permanent distortion when put under steam.

A safety valve should be placed on the exhaust end of reaction turbines to prevent any undue rise of internal pressure such as might occur were the blades stripped. The exhaust pipe should be well drained, or water may collect in it and wash back into the turbine, when in all probability the blades will be stripped. For the same reason drain pipes should connect the turbine

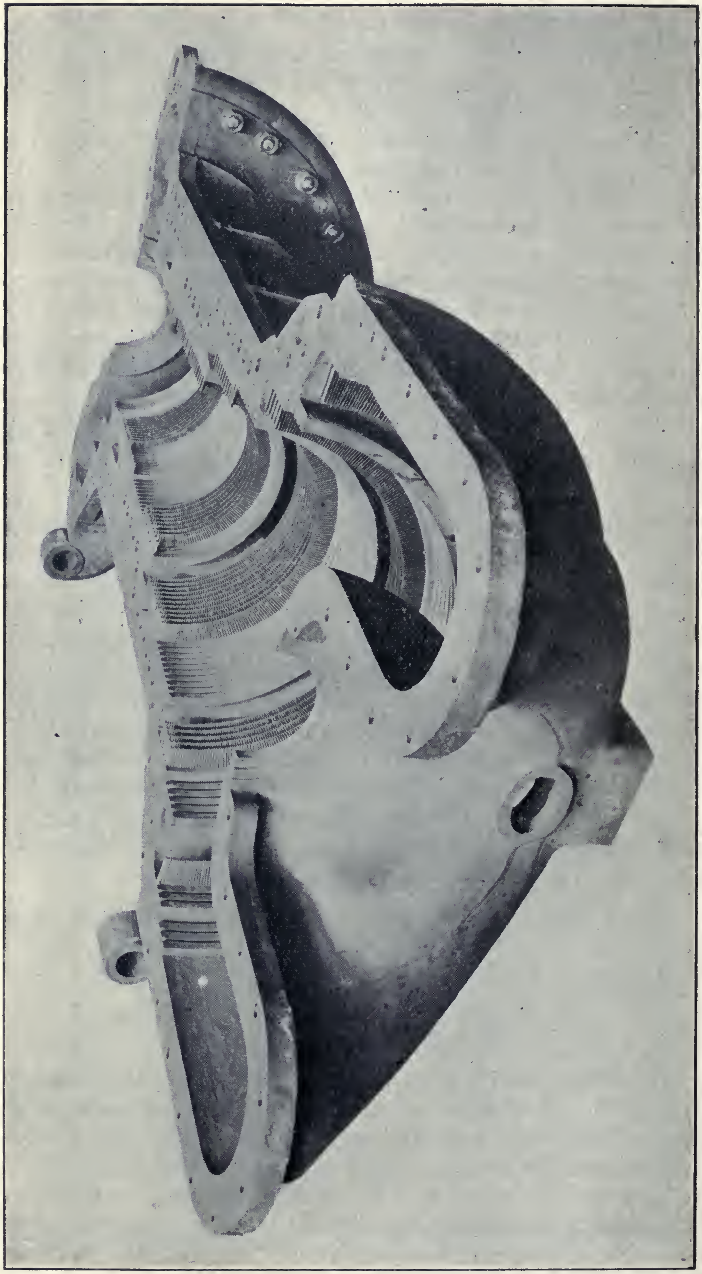


FIG. 217.—TOP PORTION OF CASING OF BRUSH TURBINE.

casing to the air pump, which latter should be started up before the turbines.

Only one end of the casing is fixed to the bedplate, usually the exhaust end. The other end is free to slide on the machined surface of the bedplate. In one impulse turbine the feet carrying the cylinder extended nearly its full length, and were provided with tongues sliding in grooved ways in the bed, in order to prevent hogging of the casing. On the other hand, feet running the whole length of the cylinder cause distortion from the circular cross-section, and are not to be recommended.

The connection between the exhaust end and the condenser must not be too rigid. An expansion piece is usually inserted. The horizontal joint between the upper and lower halves of a reaction turbine seldom has any gasket ; it is metal to metal.

The maximum tensile stress in the casing is seldom allowed to exceed 2,000lbs. per square inch under any working conditions.

Fig. 216 shows the bottom half of the casing for a Brush turbine. The two pedestals are in one piece with the main casting, thus securing certainty of alignment. Fig. 217 shows the top half of the casing. The shape of the exhaust end should be noted. The two halves of the casing are hinged to one another. Fig. 218 shows the rotor for this casing ; it is of somewhat special construction at the low-pressure end.

Fig. 219 illustrates the general constructive features of a 300 h.p. Brown-Boveri turbine. A through-going bedplate is employed, although the bearing pedestals are cast in the main cylinder body. The four guide bolts used in removing and replacing the upper half of the cylinder are also shown.

In most of these turbines the steam pipes, stop and throttle valves, and governor gear are connected to the bottom of the casing, so as to facilitate removal of the top half for inspection, and also to make the gear more getatable.

An oil tank is required with the turbine. This may be kept entirely separate, but it makes a neat construction—a little bit awkward to get at in some cases—if it is incorporated with the bed of the turbine. It is possible in this

case to arrange a through-going bed plate, which reduces the risks of cylinder distortion. The oil is cooled by means of a coil of piping carrying cold water. The well

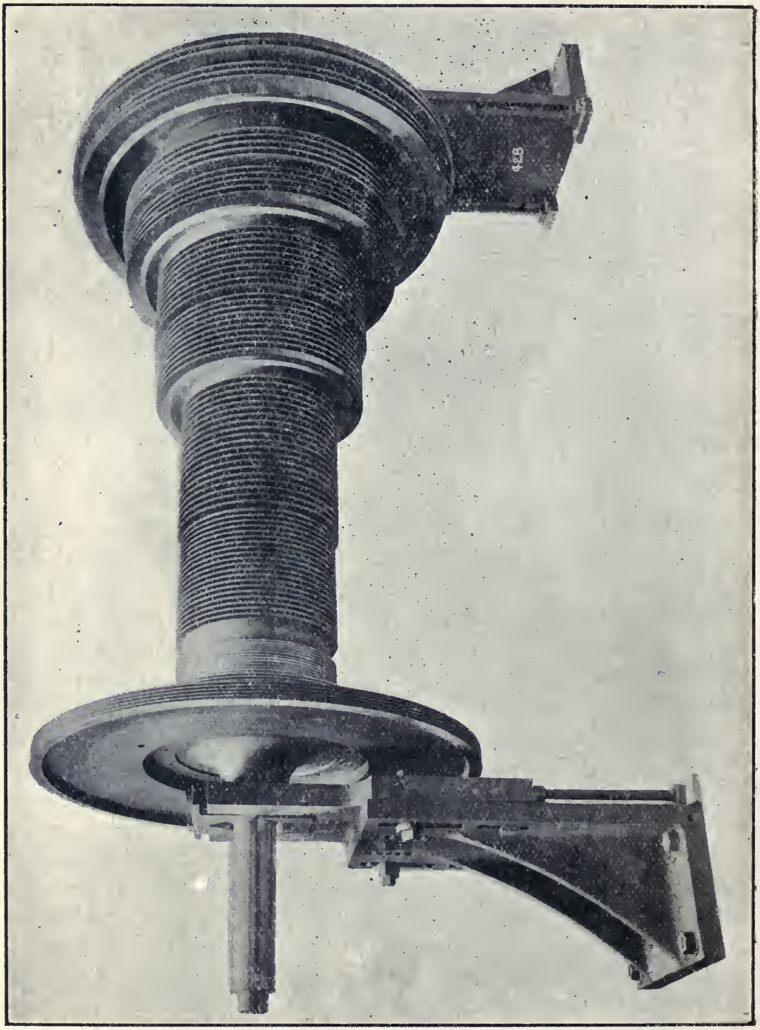


FIG. 218.—ROTOR OF BRUSH TURBINE.

from which the oil pump takes its supply should be distinct from the cooling tank and at the opposite end from the oil inlet, so as to prevent the warm oil from the bearings being taken direct to the pump without cooling.

As a rough basis for calculating the quantity of cooling water required, we may take it that the maximum quantity of heat to be removed by it does not exceed 2 or at most 3 per cent. of the maximum work done by the turbine.

Number of Stages.—The number of stages should be calculated by methods previously described, but for a rough calculation Speakman gives the rule

Square of blade velocity $\times$ number of rows $=$ constant.

This constant is about 1,500,000 for marine work and about 1,600,000 or 1,700,000 for stationary turbines. These values seem rather low for stationary turbines.

Clearances.—The clearances given below are assumed to be measured in the cold. For reaction turbines the axial clearances are, as a rule, equal to half the blade widths. The tip clearances are, of course, kept as small as possible, although it is seldom that they are less than 0·02 of an inch. No definite rule can be laid down, but as a rough rule we may say that a clearance of 0·01 of an inch is allowed for each foot of rotor diameter, with, in addition, an extra 0·01 of an inch. Another rule is to allow about 0·012 of an inch per foot of diameter. The casing is the chief obstacle to smaller clearances, on account of its distortion under steam.

In impulse turbines radial clearances are of no importance when a shroud is provided over the ends of the moving blades. The area of contact between the flowing belt of steam and the surrounding mass of steam in which the wheels rotate should be reduced to a minimum. For this reason axial clearances—more particularly at the inlet edges of the moving blades—should be kept small. The following table (Table XLIX.) gives the axial clearances recommended in Curtis turbines by a committee of the American National Electric Light Association. Baffles are frequently placed encircling the ends of the blades to restrict the influx of steam due to the suction action of the flowing belt of steam.

In general the axial clearance is limited by accuracy of workmanship and by the longitudinal expansions of the rotor and casing. Where a continuous shroud or lacing strip is used, however, another factor has to be taken into consideration. Suppose the casing to be of cast iron and the shroud of brass. Then, when under steam, the shroud or lashing will expand more than the casing and

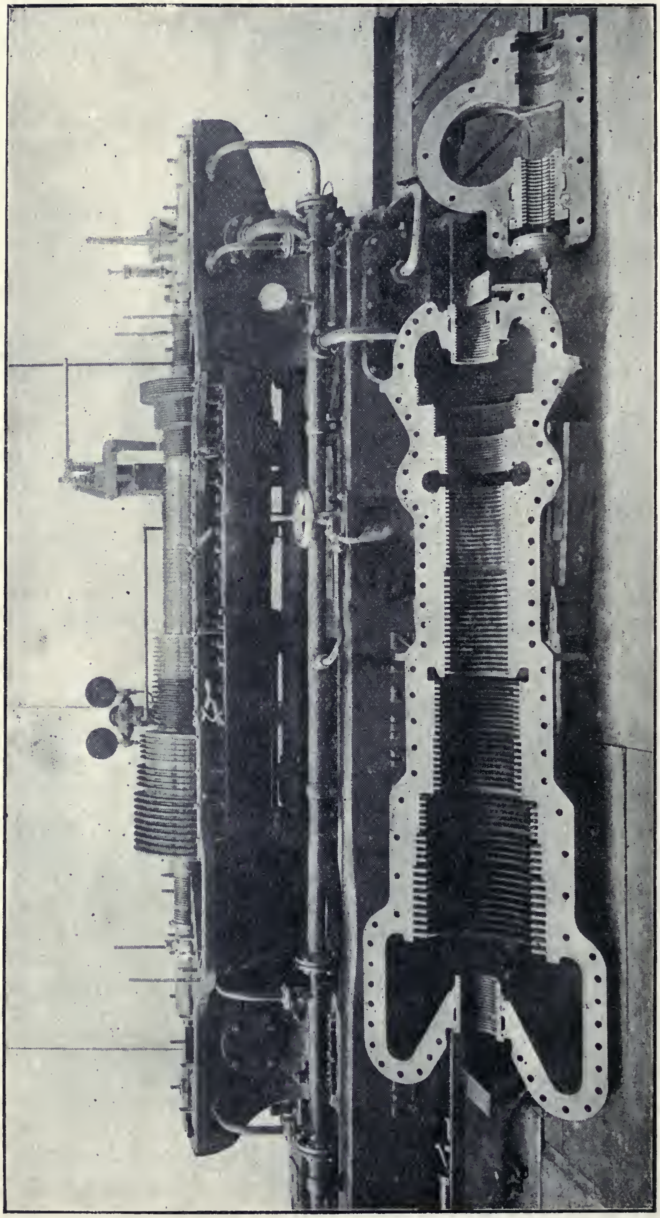


FIG. 219.—300 H.P. BROWN-BOVERI TURBINE.

will buckle. This may be obviated by making the shroud of steel or in short sections.

TABLE XLIX.

Size, Kilowatts.	Stages.	Clearance, in Inches.			
		First Stage.	Second Stage.	Third Stage.	Fourth Stage.
500	4	·06	·06	·06	·06
800	4	·07	·07	·07	·07
1,000	7	·08	·08	·08	·15
1,500	4	·06	·06	·06	·08
2,000	4	·06	·06	·08	·08
3,000	4	·07	·07	·07	·08
5,000	4	·07	·07	·07	·08
5,000	6	·10	·10	·10	·20

In view of the application of these shroudings in turbines of the Parsons type (Fig. 220) and the stripping of blades due to this buckling action, it is desirable that we should

FIG. 220.—BLADING OF WILLANS & ROBINSON TURBINE, SHOWING SHROUDS OVER BLADE TIPS.

look into the matter a little more closely. A very accurate calculation is quite impossible, in view of the difficulty of estimating precisely the conditions. The following method is only roughly approximate.

Suppose the casing in which the blade roots are to be of cast iron and the shroud of brass. Under running conditions their temperature is T° Fah. above that of erection. Consequently they expand to (initial length unity)

$$1 + \cdot000006\ T,$$
$$\text{and}\quad 1 + \cdot0000103\ T.$$

Suppose, also, that the shroud buckles into a wavy curve made up of arcs of circles, each subtending an angle θ at the centre of the circle, and having a chord or arc of span L inches. Geometrical calculation gives us the values of θ for different temperature rises T. The values are approximately as follows:—

T° Fah.	100	200	300	400	500
θ { degrees ...	6·8	7·7	8·7	9·6	10·6
θ { radians ...	·119	·134	·152	·168	·185

θ is also the slope of the shroud to the original (straight) position at the junction of two arcs.

Now, the bending of the shroud is resisted by the blades. The shroud pulls them out of the radial position with a force Fu where u is the amount (measured across the plane of the blades) by which the blade tip is pulled out of the perpendicular in inches. Assume that the blade is held rigidly at its base (a doubtful assumption), then if

N = number of blades per inch of circumference,
h = height of blade to shroud in inches,
E = modulus of elasticity of blade, pounds per square inch,
I = moment of inertia of blade cross-section about axis parallel to the plane of blading,

we have

$$F = \frac{6\,E\,I}{h^3}.$$

For example, if

$$E = 13{,}000{,}000$$
$$I = 0{\cdot}0018$$
$$h = 10$$

Then

$$F = 140$$

Consider each arc of the buckled shroud as a beam, having a maximum deflection u and a slope at the ends θ

radians, then in view of the nature of the loading, we shall not be very far out if we put

$$\theta = \frac{W L^2}{20 \, e \, i} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

and

$$u = \frac{W L^3}{60 \, e \, i} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

e = modulus of elasticity of shroud.
i = moment of inertia of shroud cross-section.
W = total pull of blades on shroud per arc of length L.
$= \frac{2}{3} F u N L$ . . . . roughly.

Inserting these values in equations 1 and 2, we see that

$$\theta = \frac{F u N L^3}{30 \, e \, i} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

and

$$1 = \frac{F N L^4}{90 \, e \, i} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

So that

$$\frac{L}{u} = \frac{3}{\theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

For example, if with the blades previously mentioned the temperature rise is 500° Fah., and if

$$N = 2{\cdot}2,$$
$$e = 13{,}000{,}000,$$
$$i = 0{\cdot}002.$$

Then from equation 4 we see that L is 9·3in. Hence, since θ is 0·185 radians, we see from equation 5 that the maximum distortion (axial) of the blading is about 0·57in.

Whilst this is an extreme case, as we seldom get long blades, except at the low-pressure end, where temperature rises are small, yet it will serve to show the danger which may arise from this buckling. Even were the temperature rise only 100°, the distortion would amount to 0·37in.

Also, we must remember that the rotor blading will also be distorted—although not quite to the same extent—so that in the latter case (100° Fah. rise) the clearance necessary to allow for distortion alone would still amount to more than half an inch.

The above calculations are only very rough—greater refinements could easily be introduced—and perhaps give rather exaggerated results. The conditions are too uncertain to allow of very precise figures, and in any case the above investigation will tell us how to reduce the distortion by buckling.

Long and weak—narrow and thin—blades increase the distortion, as does also a stiff shroud.

If a steel shroud were used with, say, brass blades, there would be an axial bending of the blades—but no buckling—due to the expansion of the brass blades being greater than that of the shroud. This can be very easily estimated, but it is almost never of any consequence, certainly not in any turbine yet built, assuming such a shroud to be fitted.

Lifting of Brass Caulking Pieces.—This unequal expansion must also be considered as to its effect on the brass foundation ring or caulking pieces in the grooves of the rotor or casing. If the caulking has not been very well done, then it is quite possible that the extra expansion of the brass may cause it to lift out of the groove. The action of centrifugal force will assist this action in the case of the rotor.

Even where there is no initial lifting, this unequal expansion may in some cases result ultimately in blade stripping. The expansion is prevented by a compressive stress in the brass. Provided this stress is small, it will serve to keep the blades in position.

If, however, this stress is large it may stress the brass beyond the elastic limit, so that by repeated coolings and heatings the caulking pieces may become slightly loose. The compressive stress is easily calculated.

T = rise of temperature above erection, deg. Fah.
E = modulus of elasticity.
= 5,500 tons per square inch.
f = comprehensive stress induced (tons).

Then

$$f = \cdot 0000037\, T\, E \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad \text{rotor.}$$

Or

$$\cdot 0000043\, T\, E \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \quad \text{casing.}$$

Thus, if the rise of temperature is 500°, the stress induced (casing) is about 9·7 tons per square inch. These stresses are of course less when using saturated steam, and towards the low-pressure end of the turbine. Steel caulking pieces have been substituted for brass in some

instances, but are difficult to caulk satisfactorily, being too hard. This objection is only partially surmounted by making these pieces of steel wider at the bottom (of the groove) than the top, previous to insertion in the grooves.

Thrust Block.—The thrust block is only a small affair, as it has only to take up a small amount of unbalanced thrust. Its main function, however, is to provide a means of adjusting the axial clearance between the rotor and casing blades. For this reason it is made in two halves adjustable in a longitudinal direction. They are so adjusted that the clearance is correct and the end play of the rotor reduced to practically nothing. The collars on the spindle are usually turned directly in the shaft, whilst those of the bearing itself consist of brass half-rings fitted into circular grooves in the bearing blocks. The thrust block should be close up to the balance pistons so as to reduce the relative movement between the casing and pistons

Couplings.—It is usual to connect the generator and turbine shafts by means of a flexible coupling. At the

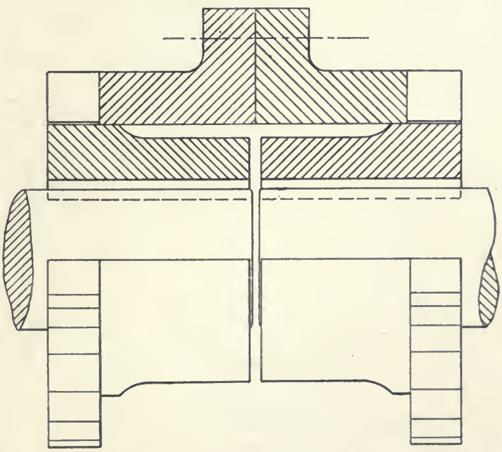


FIG. 221.—FLEXIBLE COUPLING.

same time, it is highly desirable that the two shafts should be as nearly in line as it is possible to erect them. Leather pads and drag links have been used, but it would seem that some metal-to-metal coupling is to be preferred. The main idea which lies at the basis of most of these

designs is that of providing a certain amount of play between the connecting parts, but at the same time to so arrange the shapes of the surfaces in contact that these shall always maintain contact and thus prevent knocking.

The usual type of flexible coupling used with Parsons turbines is illustrated in Fig. 221. On the end of each shaft a sleeve is keyed on. This key is provided with radial teeth or claws which engage with corresponding teeth on a second and outer sleeve connecting the two inner sleeves. This outer sleeve is made in two parts rigidly bolted together, so that by taking out the bolts the two halves of the coupling can be disconnected and either the turbine or the generator rotor removed without disturbing the other.

In view of the very slight allowance for non-alignment which these couplings are intended to provide for, a very fair coupling can be obtained if the flanks of the teeth or claws are merely radial, especially if the coupling runs

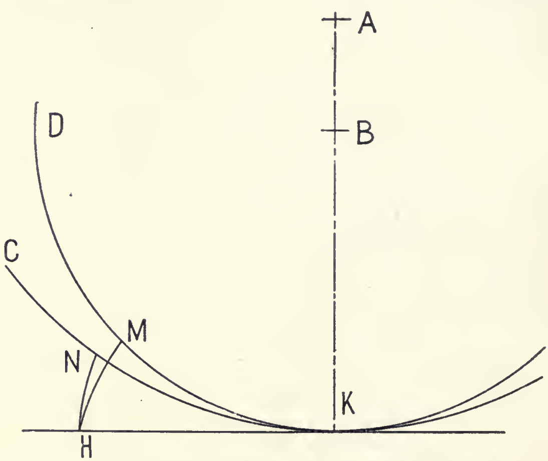


FIG. 222.—DESIGN OF TEETH FOR FLEXIBLE COUPLING.

in an oil bath. For the best results, however, the teeth should be designed as gear-wheel teeth. The shapes given to the teeth may be involute or cycloidal as in ordinary gears. Thus in Fig. 222 the circle $D\,M\,K$ represents pitch circle of one of the inner sleeves with B as centre—also the axis of the shaft—whilst the circle $C\,N\,K$ represents the pitch circle of the corresponding

outer sleeve, A being its centre, not the axis of the other shaft. Suppose we decide on involute teeth, then $H\ K$ will be the path of the point of contact between the two sleeves. If at a certain instant the position of the point of contact on this path is at H, then we see that the involutes $H\ N$ and $H\ M$ to the circles $D\ M\ K$ and $C\ N\ K$ give us the profiles of the teeth. N and M are, of course, the new positions of the points on the pitch circles which met the point H at the contact point K, the arcs $M\ K$ and $N\ K$ being equal to the distance $H\ K$.

A little consideration will show us that if the inner sleeve turns at a uniform rate the outer one will do so also, although in view of the fact that the outer sleeve is merely an intermediary, it would not matter if the transmission were not uniform.

The difference in size of the pitch circles of the inner and outer sleeves must, of course, be quite small. The distance $A\ B$ is half the maximum distance allowable between the generator and turbine shafts, and is seldom more than about a sixteenth of an inch, generally less.

In the coupling illustrated in Fig. 221 the inner sleeves are turned down between the claws, save for a narrow belt just inside the claws, so that the outer sleeve is only in contact with the inner sleeves at the teeth and these narrow belts, which thus permit of the outer sleeve taking up an inclined position relative to the shafts.

In the British Westinghouse coupling the claws or teeth extend practically the whole length of the sleeve, thus giving an increased bearing area, but making the coupling less flexible.

One very simple coupling consists in squaring the ends of the two shafts and slipping over each a sleeve with a little play, the two sleeves being firmly bolted together. If this principle were extended a little, so that on each shaft there was a sleeve with some little play, and this again enclosed by an outer sleeve with again some play, a simple and reliable coupling should result; provided that the coupling were kept flooded with oil, which, by finding its way between the sleeves, would form an elastic cushion.

One point must always be kept in mind when designing these couplings. They must always be accessible and

capable of being disconnected so that the generator, say, may be dismantled without disturbing the turbine.

Exhaust End.—In some of the smaller sizes of horizontal turbines the exhaust is led out of the casing in a horizontal direction at right angles to the shaft, below the level at which the casing is split longitudinally. Usually the exhaust passes vertically down. The exhaust end should be so proportioned as to reduce the losses between the rotor and the condenser to a minimum. There should be no sudden changes of section and no square elbows. The exhaust end proper should be so proportioned that its cross-section, taken perpendicular

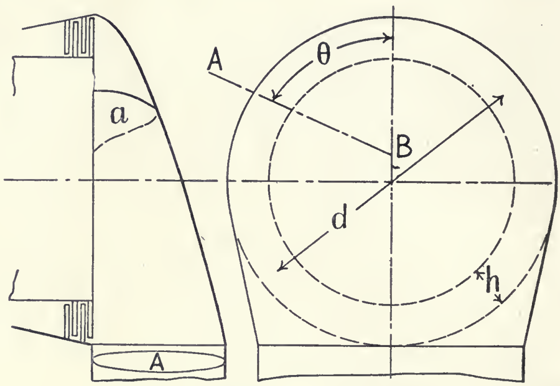


FIG. 223.—EXHAUST END OF CASING.

to the direction of flow, is proportional to the volume of steam passing. Thus in Fig. 223, if

d = mean diameter of blades at exhaust end.
h = height of last row of blades,
α = inlet angle of the blades.

Then at any section $A\ B$ distant θ degrees from the vertical centre-line, the cross-section—which is not as a rule radial in direction since the flow is not tangential in direction—on each side of the vertical will be

$$a = \frac{\theta}{360} \pi d h \sin \alpha.$$

The final section of the exhaust end will be

$$A = \pi d h \sin \alpha.$$

The reason for the presence of the factor $sin\ \alpha$ is that the absolute velocity of the steam at exhaust is greater

than its velocity parallel to the shaft in the ratio of one to *sin* a. From the turbine to the condenser the path should be as short and direct as possible. It would be advantageous to give a somewhat increasing cross-section to the exhaust pipe, as this would tend to increase the pressure of the condenser somewhat above that of the turbine exhaust. At first glance this may seem contrary to the results of Stodola's experiments on re-converging nozzles (Fig. 73), but we must remember that the velocity of the exhaust steam will seldom exceed 600ft. per second, which with an exhaust pressure of, say, 1lb. absolute and a dryness fraction of $0 \cdot 9$, would be the velocity acquired in a nozzle whose initial pressure was about $1 \cdot 14$lbs. We see then that this turbine exhaust corresponds to the case where the pressure in the nozzle is greater than that at the least section, and hence, as was pointed out when discussing Stodola's results (Curve I.), a decreasing cross-sectional area will be accompanied by a falling pressure, and conversely an increasing cross-section will be accompanied by an increasing pressure, which is what we are aiming at. That portion of the exhaust pipe with an increasing cross-section should be designed as a compressing nozzle after the manner indicated in a previous article, an allowance of 20 per cent. or even more being made for friction losses. The final velocity of the steam as it enters the condenser should be about half the velocity with which it leaves the turbine proper. To illustrate this, we will assume an example as follows :—

Velocity at rotor exhaust $=700$ft. per second ; velocity at condenser$=300$ft. per second; thermal value of kinetic energy lost $=8 \cdot 04$ B.Th.U. Of this assume that $3 \cdot 04$, or 38 per cent., is converted into heat by friction, the rest being converted into pressure energy.

Pressure at exhaust $=1$lb. absolute ; dryness $=0 \cdot 9$.

Then the pressure at the condenser will be $1 \cdot 1$lbs. per square inch, and the volume of the steam, in cubic feet per pound, will have fallen from 298 to 275. The relative cross-sectional areas must be proportional to the volumes divided by the velocities ; that is, the section at the condenser should be $2 \cdot 53$ times the section at the turbine exhaust.

The volume of the condenser gases, and hence the volume of the dry-air pump, will have been reduced about 9 per cent. by the increase in the condenser pressure, and the condenser temperature raised 3°.

The casing at the balance piston end should be designed to pass from 5 to 10 per cent. of the main steam, according to the size of the turbine. Some increase in area, above that obtained by the calculation, should be given to the exhaust end of the casing itself, to allow of a from 5 to 10 per cent. friction loss before the exhaust pipe is reached.

It may be as well to note here that the exhaust end of the rotor should be covered over by an end plate, as otherwise the inside of the rotor will act as a kind of "backwater" for the exhaust steam and will considerably reduce its velocity, without, however, increasing its pressure, and therefore requiring either a larger exhaust pipe or a decrease in the condenser pressure.

Packings.—The ordinary piston and valve-rod packings are seldom used in a turbine for the packings where the shaft leaves the cylinder. The reasons are the difficulty of keeping the ordinary packing cool and steam and air tight, without undue friction loss. What are often spoken of as labyrinth packings are generally used.

The principle of these packings is that of providing a narrow, tortuous path by which the steam may escape or air be admitted. The form of this path is arranged with a view to making this leakage a minimum.

In general such packings consist of a grooved bush enclosing a correspondingly grooved portion of the shaft. The rings of the one fit into the grooves of the other without, however, touching one another, although of course the clearances are made as small as possible.

In one form of packing (Fig. 224) the radial parts of the path, by which the steam escapes from or by which air is drawn into the cylinder, are wide when the steam (or air) flows radially outward and narrow when it flows inward. The object of this arrangement is twofold. The steam is alternately expanded and then brought to rest, so that its pressure is rapidly reduced and its power to leak lessened. Also when the steam is flowing outward owing to the wide path its velocity will be small, so that centrifugal force will not greatly assist it, whereas during

an inward motion centrifugal force will be somewhat greater and will resist the flow. It is doubtful if this latter action occurs to any considerable extent.

It will be observed that for a packing of the type illustrated in Fig. 224, whether with passages alternately narrow and wide or not, the length of the path between

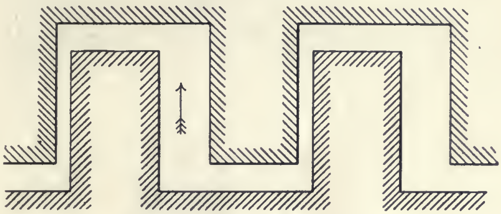


FIG. 224.—LABYRINTH PACKING.

the two ends of the packing is not greatly influenced by the size of the grooves, but that the mean cross-section of the path is somewhat greater the larger the grooves and rings.

Owing to the relative axial expansion of the rotor and casing, the clearances between the rings and grooves of these packings have to be quite large, except at the thrust-block end. In order to avoid this, the bush or the sleeve on the shaft—usually the grooves are cut directly in the shaft—may be allowed a little axial movement. This may be accomplished in several ways, one of which is to leave the bush free during warming up, and then lock it in position by means of nuts and bolts after adjusting its position relative to some mark on the rotor. Or one of the rings may be a little thicker than the others, so that when relative expansion takes place this ring comes into contact with one side of the groove in which it works and pushes the bush before it. There is then frictional contact at one ring only. This latter method is, however, hardly so satisfactory as the former. Another way of allowing for the relative expansion is by arranging the rings and grooves in concentric circles as illustrated in Fig. 225. This last method is not very often used, as it is in the way when dismantling the turbine.

A steam seal is commonly used for these packings where the shaft leaves the casing, if the steam pressure

on the inside end of the packing is below atmospheric. The steam is taken from the casing or directly from the steam pipe, and throttled to a pound or two above atmospheric pressure. It is led into the packing at about one-third of its length from the atmosphere. Thus no air will leak in, but a little steam will leak into the casing and away to the condenser. Water is sometimes used instead of steam, the American Westinghouse Company using a small centrifugal pump mounted directly on the shaft, and placed in the packing itself. This little pump

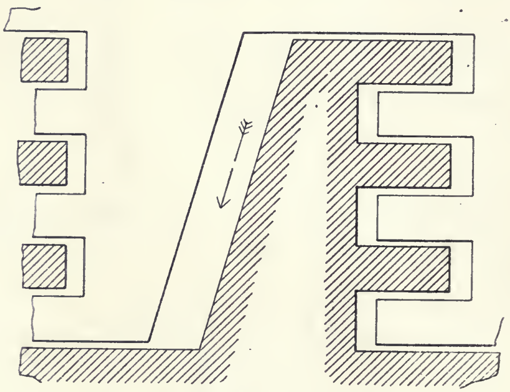


FIG. 225.—LABYRINTH PACKING.

runner forces water into the grooves of the packing on either side of it. Any water leaking out of the turbine is caught and drained away.

When running non-condensing the steam would leak out of the turbine unless a water seal, or some other means than a steam seal, were used. It is usual, when using a steam seal under such conditions, to provide the packing with a small ejector. This ejector consists of a small steam jet which sucks the steam out of the packing at the point where, when running condensing, steam is admitted. The ejector discharges into the exhaust pipe. A water seal is usually best for a non-condensing turbine.

The number of grooves used on such packings varies according to the designer's fancy ; the more the better. For reaction turbines, with the atmosphere on one side and condenser pressure on the other, the number of

grooves usually varies from 12 to 24, about one-third of them being between the steam or water seal and the atmosphere; the others on the condenser side. The bushes are usually of brass or gun-metal in halves, the halves being

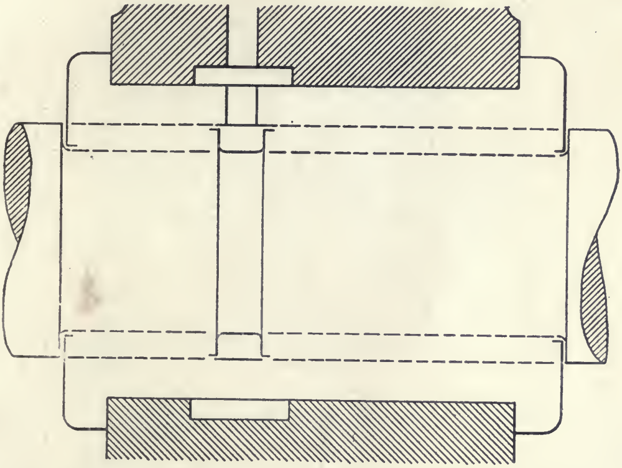


FIG. 226.—GENERAL ARRANGEMENT OF PACKING IN TURBINE CASING.

fixed—usually by set screws—to the casing. Fig. 226 represents such a packing. Each groove is rather less than ½in. wide.

The kind of packing generally used for the diaphragms inside impulse turbines has already been described.* The radial clearances used in this case are quite small, and indeed there is practically contact in some cases. Fig. 227 illustrates such a packing as used in a Curtis turbine. The bush itself fits loosely in the diaphragm but closely encircles the shaft. The drop of pressure across such a diaphragm is very considerable.

The packings used for the cylindrical surfaces of the balance pistons in reaction turbines are very similar, in a general way, to those at the points where the shaft leaves the casing. A number of circular grooves of about ¼in. in width and rather deeper are turned in the cylindrical face of the piston. Into these grooves project thin brass rings fitted into grooves cut in the casing. The

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* Fig. 13, p. 13.

clearance (on one side) between a ring and the radial face of the grooves is reduced to the smallest possible amount, generally between 0·007in. and 0·02in. Rather more than this is allowed in marine practice. These

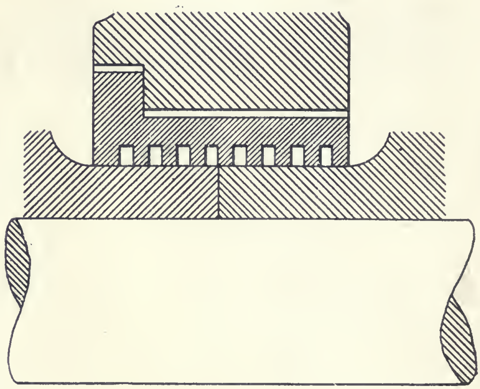

FIG. 227.—LABYRINTH PACKING IN DIAPHRAGM OF CURTIS IMPULSE TURBINE.

small clearances are only possible because the packings are so close to the thrust block, which fixes the relative axial positions of the rotor and casing. These clearances can be a little less at the low-pressure than at the high-pressure piston for the same reason.

The number of grooves used in these packings depends very greatly on the designer's fancy. For a given number of grooves and a given pressure difference, the leakage will be proportional to the diameter. The diameter increases with the power, so that the leakage does not necessarily increase—considered as a percentage of the steam consumption—if the number of grooves remains unaltered. In general the number of grooves is roughly proportional to the diameter. A rough rule for the number of grooves is as follows : The number of grooves may be made equal to 1·3 times the number of inches of diameter for the high-pressure piston, equal to two-thirds the diameter in inches for the intermediate, and equal to one-quarter the diameter in inches for the low-pressure piston. For small turbines the numbers as given by the above rule should be increased somewhat, and decreased for the larger sizes.

The brass rings used in these packings are usually about $\frac{1}{2}$in. by $\frac{1}{8}$in. in cross-section. In order to hold them in position they are sometimes slightly dovetailed, as illustrated in Fig. 228. The packings illustrated in Fig. 229

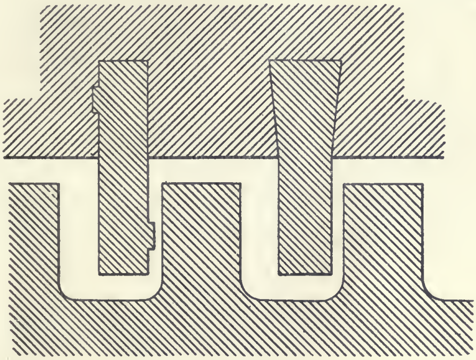

FIG. 228.—PACKINGS FOR BALANCE PISTONS.

have no such dovetailing, and depend solely on tightness of fit for maintaining them in position. The object of the projection on the packing circumferential face is to facilitate

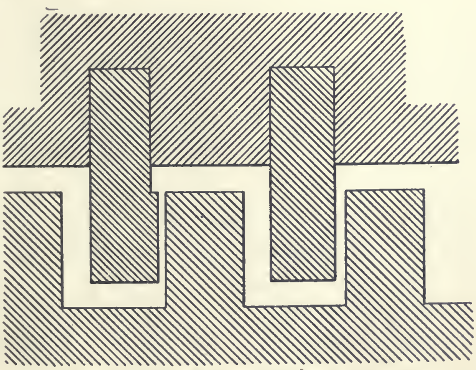

FIG. 229.—PACKINGS FOR BALANCE PISTONS.

the final turning or grinding of the rings, so that the clearance can be adjusted equally on all rings. The corners of the rings and projections on the pistons should be sharp, not rounded.

The final adjustment of these clearances is made by grinding the strips against the faces of the grooves when under steam until a good surface contact is obtained.

These packings have had to be somewhat modified for use in large marine turbines at the end away from the thrust block, owing to the large amount of relative expansion between the rotor and casing. The method adopted at first was to turn a number of grooves in the rotor and put in them bronze piston rings, usually of the Ramsbottom type. These rings pressed against the inside of the casing and it was found that undue grinding and wear took place, especially on the first few rings, which practically received most of the fall of pressure, leaving the other rings very little to do, but exposed to the débris

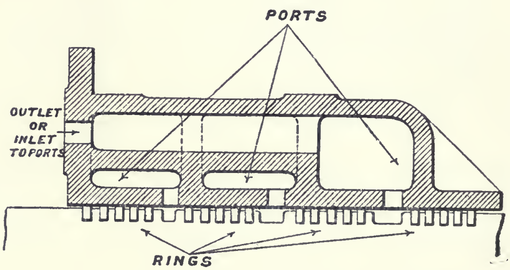

FIG. 230.—PISTON RING PACKINGS FOR MARINE TURBINES.

ground from the first rings. An attempt was made to overcome this difficulty by connecting different points on the packing to points approximately equally spaced along the turbine proper. This is illustrated by Fig. 230.

This method being still unsatisfactory, Messrs. John Brown & Co., of Clydebank, returned to a modified type of labyrinth packing illustrated in Fig. 231. In this packing axial clearances have to be large, so that steam-tightness depends on the radial tip clearances. In addition a few piston rings were added at the end of the packing as illustrated in Fig. 232, which shows the packing for the "Carmania" where the shaft emerges from the casing. The radial-fin packing illustrated in Fig. 231 had been devised by Mr. Parsons for his internal packings on the astern rotor.

Referring to the "Carmania" packing (Fig. 232) the pipe H connects to a condenser when used on a low-pressure

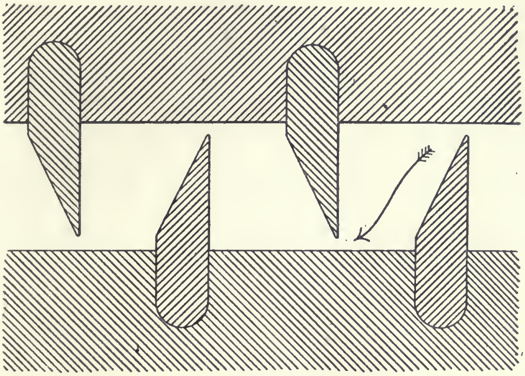

Fig. 231.—Packing for Marine Turbine.

turbine. For a high-pressure turbine the space O is generally connected to a point on the low-pressure casing

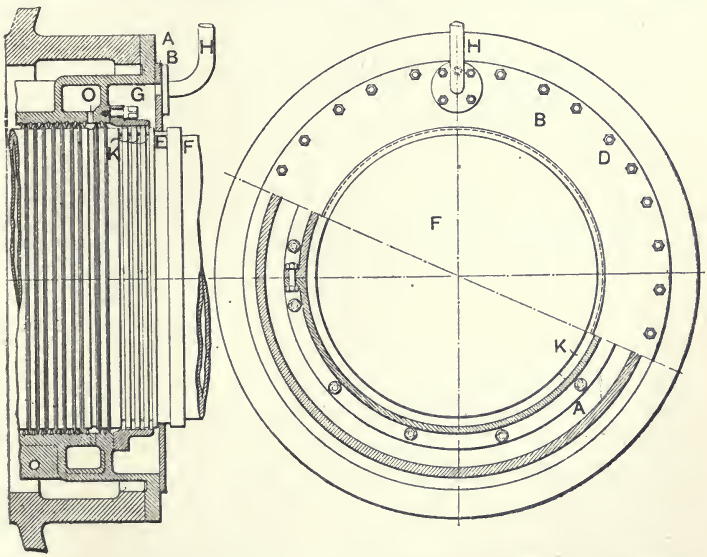

Fig. 232.—Gland for Solid Shaft through End of Casing in "Carmania."

at about atmospheric pressure. Thus there is, in any case, very little fall of pressure across the Ramsbottom rings.

In all packings placed between the atmosphere and exhaust pressure, it is good practice to arrange a pocket a few grooves from the atmospheric end which shall be kept at or just above atmospheric pressure by means of a steam or water seal. Thus there is no pressure gradient at the atmospheric end and no tendency for air to leak into the exhaust. If there is any small pressure gradient it should incline outwards.

Number of Cylinders.—It is seldom now that more than one cylinder is used in reaction turbines, except in the case of marine turbines, where the different cylinders are on separate shafts. The larger impulse turbines,

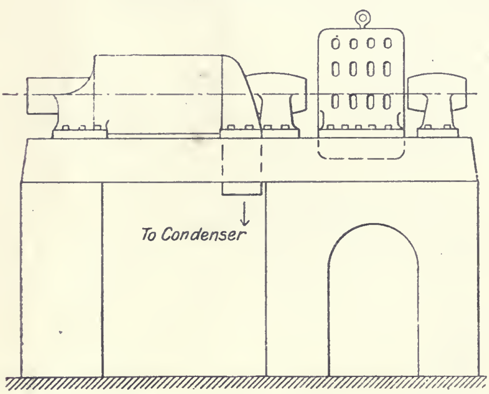

FIG. 233.—FOUNDATION FOR HORIZONTAL TURBINE.

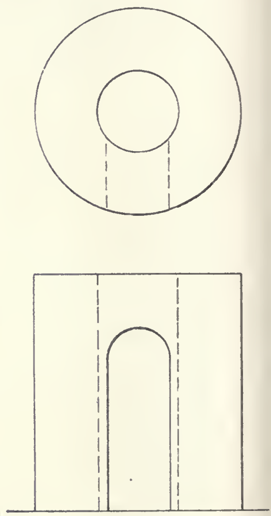

FIG. 234.—FOUNDATION FOR VERTICAL TURBINE.

however, generally have two cylinders, on account of the too great flexibility of the long shaft which would be necessary were only one cylinder used. The Curtis turbines all have one cylinder only.

Foundations.—Owing to the absence of serious vibrations, turbine foundations are comparatively cheap and simple. Vibration, however, is never entirely absent, and sometimes makes itself felt as an unpleasant tremor some distance away from the turbine. Mr. London recommends the use of layers of felt, wood, or lead, or all

three, between the turbine and its foundation in small installations. A very common foundation consists of a concrete floor carried on either a concrete arch or steel cross-beams resting on concrete side walls. The condensing equipment is then placed in the basement under the turbine. For very large units concrete pillars are sometimes placed supporting the floor under each bearing. This is illustrated by Fig. 233. The Curtis turbine is frequently supported on a hollow vertical cylinder of concrete as illustrated by Fig. 234. The footstep bearing is in the central hole, access being obtained through the doorway in the vertical face of the wall. Fig. 235 shows the foundation of a large British Westinghouse turbine. As is usual with horizontal land turbines, the condenser is placed directly under the turbine.

The simplicity of the foundation for a large turbine has been somewhat exaggerated. The main reason for the somewhat smaller cost of these foundations as compared with those of a reciprocating engine of the same size is to be found in the much smaller weights which the former have to carry. So far as simplicity of construction goes, there is not very much to choose. Except for Curtis turbines (and marine turbines of all types), foundation bolts are seldom used, although it is a common and wise precaution to embed the bed plate in at least several inches of concrete.

The foundations and general arrangement of a 1,400 kw. Willans and Robinson turbine are illustrated in Fig. 236. The position of the auxiliaries in the basement is clearly brought out.

Weights and Floor Space.—A few figures showing the weights and floor space for different sizes of turbine units are given. The floor space as usually given is merely that actually covered by the turbine, and does not make any allowance for the necessary passage-ways round the turbine or for the floor space taken up by the auxiliaries. In many installations of horizontal turbines these are under the turbine itself, in a basement; but with vertical turbines this is not often resorted to, and the auxiliaries are then responsible for a considerable proportion of the floor space. In the new power plant of the Edison Electric Illuminating Company of Boston, U.S.A., the

actual floor space covered by the 12 5,000kw. Curtis turbines—with condenser in the sub-base of each turbine—is only 6·25 per cent. of the floor space of the turbine-

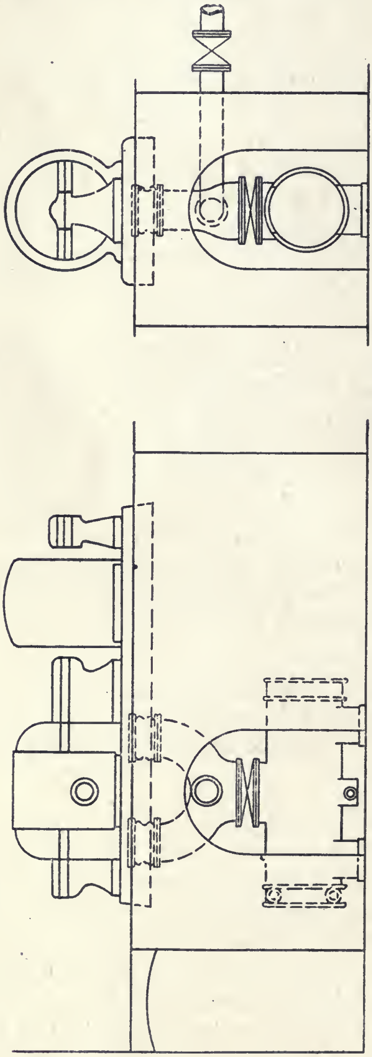

FIG. 235.—FOUNDATION FOR STEAM TURBINE.

room. The area of the turbine room per kilowatt capacity is 0·705 sq. ft., although it would not be difficult to reduce this materially.

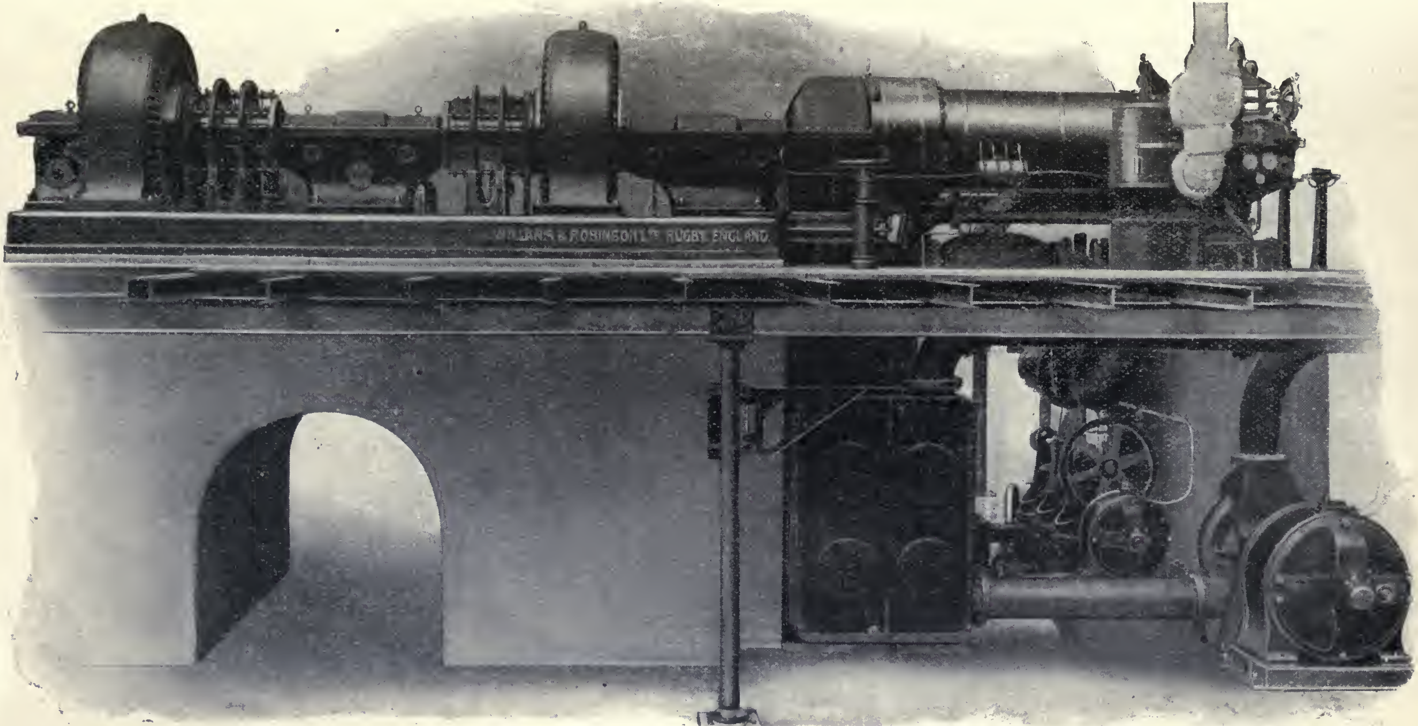

FIG. 236.—WILLANS AND ROBINSON 1,400 KW. TURBINE, SHOWING FOUNDATIONS, FLOORING, AND CONDENSING EQUIPMENT.

The accompanying table (Table L.) is from a paper by Mr. Bibbins on "Steam Turbine Power Plants," and gives the floor space and head room required for three Westinghouse designs for turbine equipments. Needless to say, the plant is as compact as it well can be, and unless space was very scarce, somewhat more room would be allowed by most engineers.

TABLE L.

Comparative Data on Turbine Plant Arrangements.

Normal capacity of units .. kw.	400	1,000	5,500
Number of units	4	4	4
Capacity of room .. .. kw.	1,600	4,000	22,000
Size of engine-room	26′ × 35′	59′ × 36′	100′ × 61′
Length of turbine units (over all) ..	18′ 11″	29′ 11″	47′ 3″
Width of turbine units (over all) ..	3′ 11″	5′ 3″	14′ 0″
Height of turbine units (over all) ..	7′ 6″	8′ 4″	14′ 0″
Centre to centre distance between units	7′ 10″	13′ 0″	22′ 6″
Width of passage-ways	4′ 0″	8′ 6″	8′ 6″
Depth of basement	14′ 6″	18′ 0″	25′ 0″
Vacuum	28″	28″	28″
Condenser cooling surface	7,000′	16,000′	80,000′
Condenser cooling surface, per unit	1,750′	4,000′	20,000′
Condenser cooling surface, per kw...	4·37′	4·00′	3·14′
Area of operating-room .. sq. ft.	910	2,124	6,100
Turbine capacity, per square foot of operating-room .. .. kw.	1·76	1·88	3·60
Area of engine-room, per kilowatt capacity	0·57	0·531	0·277
Area of engine-room, per E.H.P. capacity	·425	·396	·207

It is interesting to note how the floor space per kilowatt capacity decreases with the size of the units. The condensing plant is in all cases in the basement under the turbine.

Below are some figures by W. F. Wells, showing the power-house and engine-room floor spaces for the five great New York power stations.

	Area—sq. ft. per kilowatt capacity. Power House.	Operating Floor.
New Edison Company	0·96	0·573
Metropolitan Railway	1·27	0·635
Kingsbridge Station	1·40	0·748
Manhattan Railway	2·06	0·884
Rapid Transit	2·32	1·38

The most compact of the five is the New York Edison station. At the time the above figures were computed

it contained only vertical three-cylinder engines of 6,500 h.p. each. Some 5,000 kw. Curtis turbines are now being added. The Rapid Transit and Manhattan stations have double horizontal-vertical engines of about 10,000 h.p. each. The Metropolitan and Kingsbridge stations have vertical cross-compound engines of 4,500 h.p. each. In three—if not all five—the auxiliaries are in the basement between the concrete foundations. Fig. 237 is taken from the paper by Mr. Bibbins, and illustrates the floor space covered by various types of prime mover of

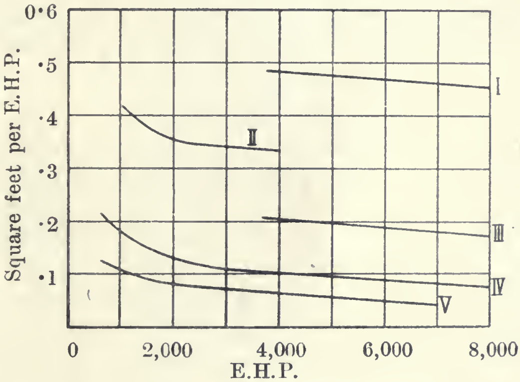

FIG. 237.—FLOOR SPACE OF NEW YORK POWER STATIONS.

I.—Horizontal-vertical: Manhattan.
II.—Vertical Cross Compound Corliss: Kingsbridge.
III.—Vertical 3-cylinder Compound Corliss: New York Edison.
IV.—Westinghouse Turbine.
V.—Curtis Turbine.

the rated capacities indicated. The present writer has added a curve showing the floor space covered by Curtis turbines. Auxiliaries are not included in the above figures. It will be seen that the Curtis turbine by itself occupies less floor space than any of the others. As regards the height, some figures have already been given in the table by Bibbins. A Curtis turbine of 5,000 kw. turbine stands 23ft. 6in. above its base (without condenser) and a 500 kw. turbine 12ft. 5in.

In Table LI. are given the overall dimensions of Parsons turbines as made by a British firm. The dimensions do not include anything for passage-ways or condensing plant.

TABLE LI.

Kilowatts	75	150	250	500	1,000	1,500	2,000
Revs. per minute ..	4,000	3,500	3,000	2,500	1,500	1,500	1,200
Length .. .. ft.	10·5	12·1	14·25	17·6	19·4	20·6	20·6
Width .. .. ft.	2·5	3·25	3·5	6·75	8·0	9·5	9·5
Height .. .. ft.	5·0	6·0	7·0	7·0	9·5	9·5	10·0
Floor space	26·2	39·3	50	119	155	196	196
Square feet per kilowatt	0·35	0·26	0·20	0·24	0·16	0·13	0·10

It will be noticed how the floor space per kilowatt decreases as the size of the turbine increases.

In Table LII. are given the dimensions for the complete turbo-generating sets—without auxiliaries.

TABLE LII.

Kilowatts	75	150	250	500	1,000	1,500	2,000
Length	16·5	20·5	22·5	25·5	37·0	42·0	50·0
Width	2·5	3·25	3·5	7·0	8·0	9·5	10·5
Height	5·0	6·0	7·0	7·0	9·5	9·5	10·0
Square feet per kilowatt	0·55	0·45	0·32	0·36	0·30	0·26	0·26

In Table LIII. are given some particulars as to the weights of various turbines. In Table LIV. are given some data respecting steam reciprocating sets, whilst in Table LV. we have some data as to gas-driven sets. As regards the reciprocating engines, a good deal depends on the type of engine and something also on the specifications of the engineer for whom they are made. The large 4,000 kw. gas engine is for driving a Crocker-Wheeler generator in the power-house of the California Gas and Electric Company. It is intended for intermittent use, which may partly account for its relatively small weight.

It will be noted that the American Westinghouse turbine seems to be about 40 per cent. heavier than others of a similar type. The author believes that this is largely due to the extremely liberal rating accorded these machines, most of them carrying about 25 per cent. overload before opening the by-pass valve.

Turbines for Central Station Work.—As we have already pointed out, the steam turbine stands about level with the reciprocating engine so far as steam consumption and oil and condensing water combined are

TABLE LIII.—*Weights of Steam Turbines.*

Turbine.	Rated Capacity Kw.	Weight in Tons.			Weight in Pounds per Kilowatt.	
		Whole Unit.	Turbine only.	Turbine Rotor.	Unit.	Turbine.
Westinghouse-Parsons ..	5,500	250	157	..	102	64
	1,500	78	..	12·5	117	..
C.A. Parsons ..	4,000	110	..	12	62	..
,, ..	3,500	93	..	..	60	..
Brush-Parsons ..	5,000	..	42	..	..	19
,, ..	3,000	..	32	..	..	24
,, ..	2,000	60	25	..	67	28
,, ..	1,000	43	14	2·8	96	31
,, ..	500	19	9	..	85	40
Brown-Boveri-Parsons ..	5,000	150	..	..	67	..
Curtis	8,000	312	..	..	88	..
,, ...	5,000	180	81	62·5*	80	36
,, ...	3,000	110	75	..	82	56
,, ...	2,000	85	51	..	95	57
,, ...	1,500	56	36	..	83	54
,, ...	500	18	12	..	80	54
British Westinghouse	5,500	170	80	20	69	33

* Including generator field.

concerned. As to whether there is any saving in attendance costs is open to question. In one power station the author is acquainted with, there is is an engine driver continuously attending each turbine. This policy is not followed in most power-houses. A good deal of the attendance given to generating sets is given not to the main engine but to the condensing plant, and this is at least as troublesome when turbines are employed as when using reciprocators. In these days of self-oiling engines there is not much gain as regards the greasers in attendance.

Where the turbine shows up to best advantage is in overload capacity, the reduced capital cost of the power

Table LIV.—*Weights of Steam Reciprocators.*

Engine.	Rated Capacity Kw.	Weight in Tons.			Weight in Pounds per Kilowatt.	
		Whole Unit.	Engine only.	Fly-wheel.	Unit.	Engine.
Brush high-speed	2,500	..	125	..	..	..
engine	2,000	..	120	..	..	..
,, ,,	1,000	105	60	..	235	134
,, ,,	500	47	30	..	210	134
Bellis and Morcom	800	71·5	41·5	..	200	116
high-speed engine	250	..	14	..	..	125
Sulzer slow-speed	4,000	647	450	..	360	250
engine	1,600	301	170	..	420	240
Musgrave slow-speed engine	2,500	..	750	145	..	675
Allis-Chalmers ..	2,500	..	700	105	..	625

Table. LV.—*Weights of Gas-driven Generating Sets.*

Gas Engine.	Rated Capacity in Kw.	Weight in Tons.			Weight in Pounds per Kilowatt.	
		Whole Unit.	Engine Only.	Fly-wheel.	Unit.	Engine.
Cockerill	400	..	125	32·5	..	700
Deutz	650	..	219	19	..	750
,, ...	400	..	98	12	..	550
Körting	700	..	190	45	..	610
,, ...	400	..	90	15	..	500
Simplex	400	..	125	32·5	..	700
For San Francisco ..	4,000	600	..	..	336	..

station, and the cheaper generator which can be used with the turbine.

A turbine which is designed to give maximum economy at its rated full load, can usually be arranged at very little extra cost, to give from 50 per cent. to 100 per

cent. increased load for any length of time. It is true the economy falls off somewhat, but not so rapidly as to constitute a serious objection. The gain in capital cost is considerable, although, of course, it is mainly confined to the turbine itself and—to some extent—to the buildings, which by the use of a smaller turbine are thus of somewhat smaller size than would otherwise be the case. On the other hand nothing is saved on the electrical side of the plant, and the boilers have to be perhaps a trifle larger on account of the reduced turbine efficiency at overloads.

The capabilities of the turbine itself in the way of accommodating large overloads with the aid of a steam by-pass, are especially valuable in traction stations. Large overloads lasting for a minute or two, which cannot be met by the ordinary reciprocator, are easily taken care of by the turbine; and in view of the momentary character of these heavy overloads, no damage is done to the generator.

The turbine itself costs about as much as the reciprocator except for the small and medium sizes, when it costs more. Exact comparisons are impossible, as anyone conversant with the many factors which determine the selling price of these large turbines and engines will be aware.

TABLE LVI.—*Costs of Turbine and Reciprocating Sets.*

Rated Capacity. Kw.	Steam Unit only.		Generator only.		Combined Unit.	
	Turbine.	Recip.	Turbine.	Recip.	Turbine.	Recip.
50	537	395	358	218	895	613
75	659	543	439	320	1,100	863
100	782	680	520	456	1,300	1,136
150	1,026	956	684	560	1,710	1,516
200	1,270	1,235	848	675	2,120	1,910
250	1,510	1,530	1,010	780	2,520	2,310
300	1,760	1,810	1,170	912	2,930	2,730
400	2,240	2,400	1,500	1,120	3,740	3,520
500	2,730	2.920	1,820	1,410	4,550	4,330
750	3,960	4,330	2,640	2,400	6,600	6,730
1,000	5,180	5,630	3,460	3,700	8,640	9,330

The above figures do not take into account the greater cost of the condensing plant required with a turbine.

It may be interesting, however, to compare the listed prices of turbines and high-speed engines as made by a large firm of British engineers. In this particular case the costs are the same for the two types at about the 300 kw. size, the turbine being more expensive in the smaller sizes. The data are presented in Table LVI.

As regards the costs of the generators. The high speeds adopted in turbine work have two distinct and opposing effects. The quantity of material, particularly cast iron and copper, is largely reduced by the use of high speeds, as is evidenced by the figures in Table LVII., which give the weights of two British Thomson-Houston 1,500 kw. generators built to the same general specification, save as to speed. The weights are given in pounds.

TABLE LVII.

	Turbo-generator.	Slow Speed.
Revolutions per minute	1,000	94
Stator complete with end shield.. ..	25,000	40,000
Rotor complete, shaft and half coupling	11,000	30,000
Flywheel	..	90,000
Alternator complete	36,000	160,000
Active material, sheet iron	16,000	16,000
Active material, copper	2,500	8,500
Flywheel effect, foot-tons	8,750	3,700

On the other hand the high speeds necessitate the greatest care in the selection of material and in the workmanship put into the machine. There is, indeed, a point beyond which any further increase in speed is only obtained at an increased cost. As a rule a large turbo-generating set costs less than a corresponding reciprocating set on account of the reduced cost of the alternator.

The turbine effects probably the greatest saving in large stations erected in crowded districts, where every foot of land counts. The reduced floor area and, to some extent, height of the engine-room, lead to considerable economies in capital outlay.

In view of the value of a high vacuum for a turbine, a plentiful supply of good condensing water is of importance,

and the lack of it may sometimes turn the scale against the adoption of turbines.

As we saw when considering the question of the best vacuum, a low vacuum is desirable when the load factor is very low or coal very cheap. Under such circumstances a reciprocating engine is generally superior to the turbine, except in very crowded districts. In such cases, however, it is conducive to all-round economy to have both turbines and reciprocators. The former should be kept as nearly as possible continuously at work with a high vacuum, and the reciprocators should assist at the peak loads only and be non-condensing.

Economic Considerations in Design.—Since the ultimate efficiency of a machine depends not only on its physical efficiency but also on such things as capital cost and maintenance, it is of the first importance that every effort consistent with sound engineering should be made to produce a cheap machine. The best turbine is not in general the one in which economy of steam consumption has been the exclusive aim of the designer. The turbine which saves £100 a year in fuel at the expense of £110 a year in increased fixed charges and maintenance is saving nothing, but losing £10 for its owner.

As an example of the subordination of extreme steam economy to simplicity of construction we have the Curtis turbine. The author considers that, thermodynamically, the Curtis turbine with two or more sets of moving blades per stage is less efficient than an impulse turbine with only one set per stage. The use of two or more sets of moving blades mounted upon one disc or wheel, however, considerably reduces the number of diaphragms and discs in the turbine, and thus tends towards a less costly turbine. For turbines running only a few hours per day and with cheap fuel this mode of construction is undoubtedly justified. Whether it is justified where fuel is dear and the load factor high is doubtful; sufficient data for a reliable judgment not being available.

Then, again, there is the question of the vacuum for which the turbine should be designed. In some cases the vacuum is specified, although even then it may not be advisable to alter a standard design to suit the special case. Generally speaking, a back pressure at the turbine

exhaust of 26in. or 27in. (barometer 30in.) should be assumed, although some firms work out their designs on the basis of a 28in. vacuum.

The question of the best peripheral speed is exceedingly important. Within limits, the higher the peripheral speed the more efficient the turbine. The most important limit to this proposition is the size of the blades, which, unless the revolutions per minute are abnormally high, soon makes itself felt. For a fixed speed of revolution it is, however, by no means always true that the highest blade speeds are the best.

The cost of the turbine goes up very rapidly with the diameter—not quite as the square of the diameter in most cases—the exact rate of increase depending on the type and size of the machine. On the other hand, the cost decreases with the length, which decreases with an increase in the diameter. The active portion of the turbine decreases very nearly inversely as the square of the diameter—not quite so rapidly in impulse turbines—but the inactive portions, such as the casing ends, balance pistons, &c., usually increase with the diameter.

Thus there will be, in most cases, some diameter giving a minimum cost of construction. This again may not be the best diameter, as the factor of efficiency has been left out of account.

The method of balancing saving in fuel against increase in fixed charges, so as to arrive at the most commercially economical arrangement, was fully described in Chapter X., and should be applied to such questions as the best initial pressure, best vacuum, best superheat, &c. The influence of load factor and the price of fuel will be found to be all-important in most cases. The average load factor for electric generating stations is only about 17 per cent., but there are grounds for hoping that this may be improved, and hence, unless the load factor can be estimated, we may assume it is somewhat higher than the above figure. The price of fuel depends on circumstances quite outside the designer's control.

Where a combination of reciprocating engines and turbines is adopted, the load factor on the turbines can easily be raised to 30 or 40 per cent., in which case a very high vacuum and (probably) high superheat would be economical.

As it is, forcing a turbine to work at very high vacuum with cheap fuel and a small load factor is putting it under disadvantages—except as regards steam consumption—as compared with its rivals, with which it ought not to be saddled. The writer is aware that ultimate commercial economy is too often made subsidiary to a good test result; and that is one reason why stress is here laid on the fact that the turbine with the smallest steam consumption is seldom the best.

It would be an advantage to the turbine industry (in the long run) if specifications were to call for tenders for two or three alternative arrangements (for complete plant); thus, for instance, complete equipments for 26in., 27in., and 28in. vacuum plants might be specified. The consulting engineer could then calculate all operating and fixed charges on the alternative schemes and choose the best. If this were done, some preconceived notions about economy would be rudely shattered.

Chapter XV.

HISTORY OF THE STEAM TURBINE.

It is not proposed here to deal at length with the history of the development of the steam turbine.

Interesting as that history is, it is a little outside the scope of this work. A brief sketch may, however, be of some interest to readers.

About 120 B.C. the Alexandrian philosopher, Hero, described a simple reaction turbine. This is illustrated in in Fig. 238. It consisted essentially of a steam boiler, or urn, supporting a hollow globe by means of trunions.

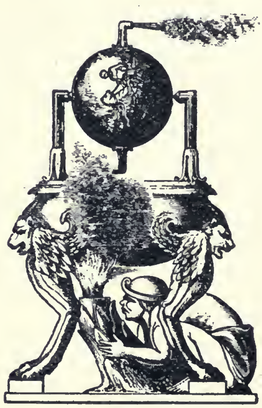

Fig. 238.—Hero's Turbine: 120 B.C.

One of these supports was hollow and supplied the globe with steam from the boiler. Two short pipes discharged the steam from the globe in opposite tangential directions, the reaction of the escaping steam causing the globe to rotate.

In 1629 Branca, an Italian, invented a kind of impulse turbine, as shown in Fig. 239. Steam from the boiler is discharged against the blades of a wheel built somewhat

like a common water-wheel. This turbine was used, or intended to be used, to drive a mortar.

Coming to more modern times, a patent was granted in 1784 to Wolfgang de Kempelen for the turbine illustrated in Fig. 240. It consists of a horizontal pipe D

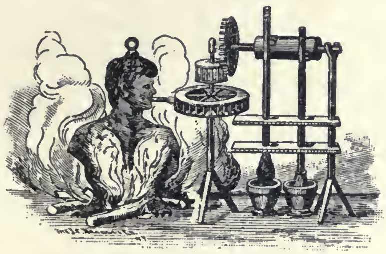

FIG. 239.—BRANCA'S TURBINE: 1629.

pivoted at E, and receiving steam from the boiler A. The steam was to be discharged from the ends of the pipe by two holes facing in opposite directions in a horizontal plane, the reaction of the escaping steam causing the pipe to rotate.

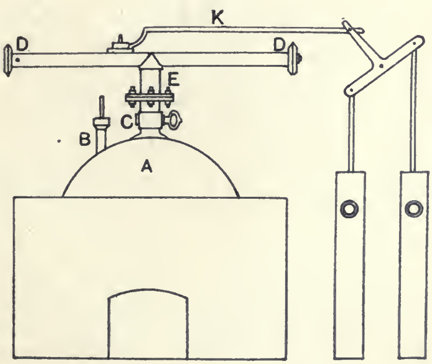

FIG. 240.—KEMPELEN'S TURBINE: 1784.

In 1784 also James Watt took out a patent for a steam turbine, but he does not appear to have proceeded with it.

Several other turbine patents were granted before 1843, when Pilbrow obtained his patent.

Pilbrow had got clear and sound conceptions as to steam turbines. He appears to have experimented with steam nozzles and carefully studied his experimental results for we find that he estimates that the best peripheral speed for his impulse wheel—on the De Laval principle—when supplied with steam at 60lbs., and running non-condensing should be about 1,250ft. per second. Now the theoretical velocity of steam issuing from a nozzle under these conditions would be 2,390ft. per second, and the most efficient blade speed about 1,200ft. per second. This fact alone marks out Pilbrow's patent as a kind of milestone in the evolution of the steam turbine.

Pilbrow proposed a turbine on the now well-known De Laval principle. In his proposed construction, however, the steam nozzles and moving blades were to be in the same plane of rotation, and blades were added to lead away the exhaust steam.

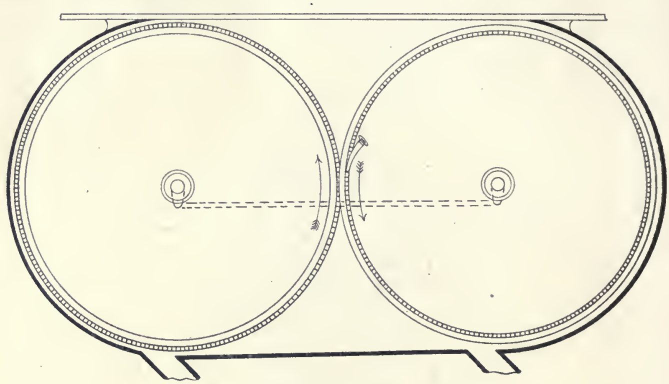

FIG. 241.—PILBROW'S TURBINE: 1843.

Pilbrow attempted to obtain a lower speed of rotation by using several wheels through which the steam passed in succession. Figs. 241 and 242 illustrate such an arrangement in which there were two parallel shafts, the wheels on which overlapped opposite the nozzle.

Pilbrow, moreover, patented the principle now used in the Curtis and Riedler-Stumpf turbines, whereby fixed blades or chambers were used to return "the steam back upon the wheel for a second or other impulses."

In 1848 Wilson, of Greenock, patented a radial flow turbine consisting of alternate rings of fixed and

moving blades (Fig. 243) and another of the "in and out" class of radial turbine (Fig. 244). Still another of his proposals was for a parallel-flow turbine. In all his proposals he made allowance for the necessary increase

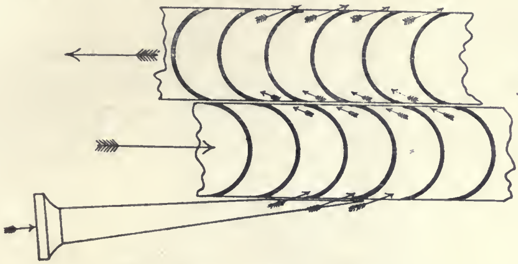

FIG. 242.—NOZZLE AND VANES. PILBROW'S TURBINE.

in the passage cross-sections as the exhaust was approached, and he undoubtedly had clear ideas on turbines for the times he lived in.

In 1875 Prof. Osborne Reynolds, of Manchester, took out a patent for improvements in fluid pumps and turbines.

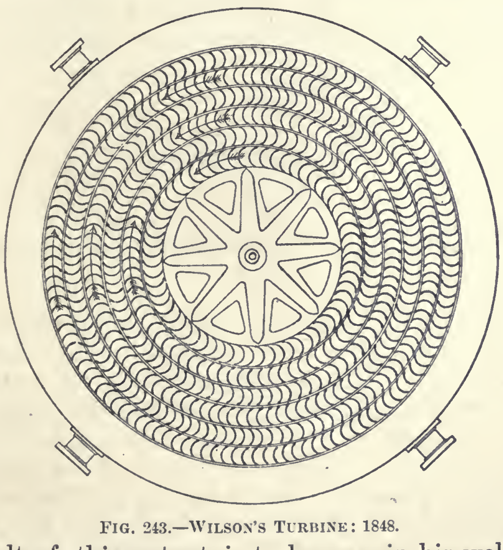

FIG. 243.—WILSON'S TURBINE: 1848.

One result of this patent is to be seen in his well-known multi-stage high-lift centrifugal pump. The patent was also intended to apply to turbines, including steam and gas turbines.

Moreover, about this time a small reaction steam turbine was made by the present writer's father for Prof. Reynolds and operated at about 12,000 revs. per minute. This was probably the first working reaction turbine, if we except modifications of Hero's well-known reaction globe.

This turbine is illustrated in Figs. 245, 246, and 247.

It was a simple radial-flow turbine, or rather, two turbines side by side. The steam flows radially outwards and the passage areas in the second stage are greater

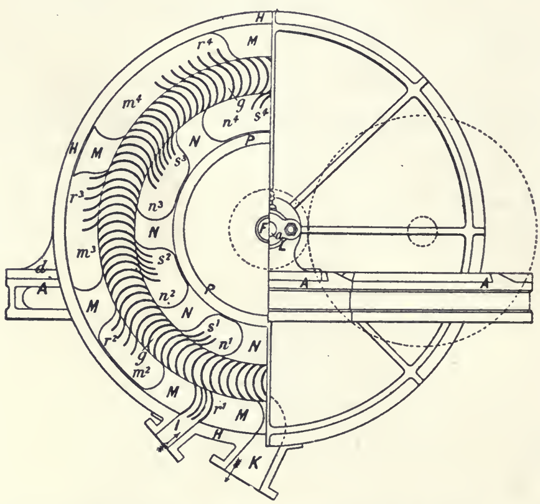

FIG. 244.—WILSON'S RADIAL FLOW TURBINE, WITH SINGLE RING OF MOVING BLADES.

than those in the first. The turbine was run non-condensing, and never took steam at more than 40lbs. per square inch. The experiments were not carried very far, because Prof. Reynolds was of the the opinion that not more than 60 or 70 per cent. of the available heat in the steam could be converted into work on the turbine shaft. This opinion has since been abundantly confirmed, but unfortunately Prof. Reynolds forgot that a condensing turbine could make good use of the toe of the indicator diagram.

The diameter of the outer rows of moving blades in the Reynolds' experimental turbine is 4·25in., giving a

peripheral speed of 220ft. per second. The outlet angle of the blades is about 20°.

In the seventies Prof. Reynolds also had a 13-stage radial flow reaction turbine, having five different diame-

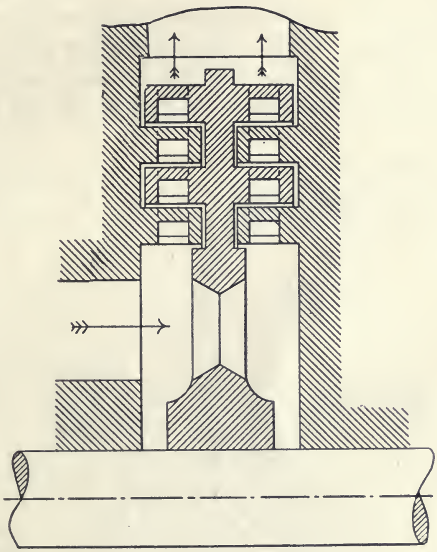

FIG. 245.—SECTION THROUGH REYNOLDS' STEAM TURBINE.

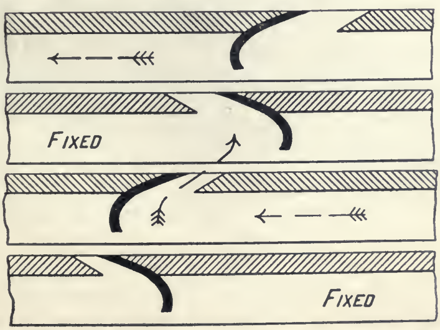

FIG. 246.—REYNOLDS' TURBINE: BLADING ARRANGEMENTS.

ters, made. The writer has been privileged to see Prof. Reynolds' calculations, which included those for a turbine using chloroform vapour.

In 1882 De Laval took out a patent for a modified Hero engine, which is illustrated in Figs. 248 and

249. This turbine was used in creameries. De Laval took out his main patent in 1889, and followed this up by several others. These patents referred to the divergent nozzle and flexible shaft in particular.

The Hon. C. A. Parsons, who has undoubtedly done more than anyone else to make the steam turbine a practical success, took out his first patent in 1884. His first turbine was double ended, the steam entering at the middle of the drum, and flowing towards the two ends.

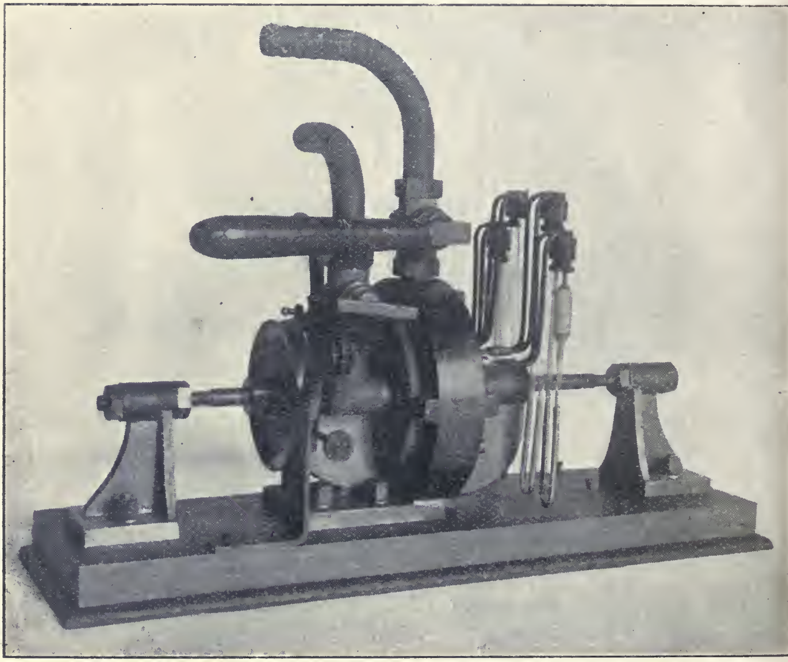

FIG. 247.—REYNOLDS' TURBINE.

Later on, influenced by patent difficulties, he developed the radial-flow turbine. The first marine turbines—those of the "Turbinia"—were radial flow. For mechanical reasons this type has been entirely superseded.

Rateau developed his turbine between 1897 and 1900; and Curtis, in America, began on his about 1895, but nothing satisfactory was accomplished until 1900.

The Curtis turbine originally had only one expansion with six or seven sets of moving blades fed from one set of nozzles. The first commercial Curtis turbines—supplied to Chicago in 1901-2—had two stages with four sets of moving blades per stage. Most of the later Curtis turbines have four stages with two sets of moving blades per stage, and a recent low-pressure Curtis turbine had only one set of moving blades per stage.

The first marine turbines were fitted in the "Turbinia" in 1895. The boat was 100ft. long by 9ft. beam, with a displacement of 44·5 tons. A speed of about 35 knots was attained. In 1905 every British warship laid down was equipped with Parsons turbines, and in 1905 the

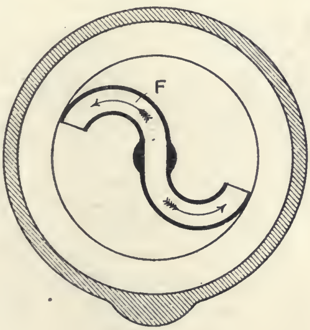

FIG. 248.

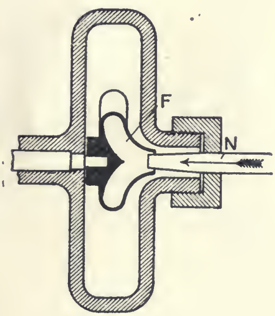

FIG. 249.

DE LAVAL TURBINE: 1882.

two first Atlantic liners using turbines—"Victorian" and "Virginian"—were put in service. They used Parsons turbines.

The Curtis, Rateau, and Zoelly turbines have also recently been fitted in ships. As pointed out in the chapter on marine turbines, impulse turbines are more particularly suitable for slow ships, although the Rateau turbine has been successfully applied to torpedo boats.*

* Those readers who are particularly interested in the history of the turbine should refer to "The Steam Turbine," by R. M. Neilson, and to a paper by the same author on "The Evolution and Prospects of the Elastic-fluid Turbine," in the Transactions of the Institution of Engineers and Shipbuilders in Scotland, vol. xlix., part iii., from which a few of the illustrations in this chapter have been obtained.

APPENDIX.

Those who have occasion to make many calculations in connection with steam turbine work will require a set of formulæ and general data. Still the following data will be found useful, especially for those who have not a complete set of data in another form :—

Table LVIII.

Coefficients of Linear Expansion per Degree Fahrenheit; and Specific Heats of Materials.

MATERIAL.	Coefficient.	Specific Heat.
Wrought iron	0·0000073	0·114
Steel	0·0000067	0·116
Cast iron	0·0000060	0·130
Brass	0·0000010	0·094
Copper..	0·0000010	0·092
Masonry and cement..	0·000008	0·2

Table LIX.

Properties of gases, see chapter XIII.

Gas.	Symbol	R	kp.	kv.	Kp	Kv.	n*	m*	C.
Air	..	53·18	·238	·169	184·5	131·4	1·407	·711	..
Oxygen	O	48·2	·218	·156	169·5	121·3	1·398	·715	..
Hydrogen	H	770	3·406	2·416	2,650	1,880	1·410	·710	61,260
Nitrogen	N	55·5	·244	·173	190	134·5	1·410	·710	..
Carbonic acid	CO_2	48	216	·154	168	120	1·403	·713	..
Carbonic oxide	CO	56	·248	176	193	137	1·410	·710	4,370
Marsh gas	CH_4	95	·593	·470	461	366	1·26	·793	26,400
Olefiant gas......	C_2H_4	..	·404	..	314	..	..	..	21,300

C=B.Th.U. liberated by combustion in oxygen.
* These values of n and m are for adiabatic expansion.

TABLE LX.

Lbs. per square in. absolute p.	Index m. Dry and Saturated Steam.	Index m. Increase per 1 per cent. moisture.	Lbs. per square in. absolute p.	Index m. Dry and Saturated Steam.	Index m. Increase per 1 per cent. moisture.
250	·8854	·00082	120	·8784	·00094
240	·8849	·00083	110	·8778	·00095
230	·8843	·00084	100	·8773	·00096
220	·8838	·00085	90	·8768	·00097
210	·8832	·00086	80	·8762	·00097
200	·8827	·00086	70	·8757	·00098
190	·8822	·00087	60	·8751	·00099
180	·8816	·00088	50	·8746	·00100
170	·8811	·00089	40	·8741	·00101
160	·8805	·00090	30	·8735	·00102
150	·8800	·00091	20	·8730	·00103
140	·8795	·00092	10	·8724	·00104
130	·8789	·00093	0	·8719	·00105

Explanation of the table :—

m is the index in the expansion curve.

$$p^m \, u = \text{a constant},$$

u being the volume of 1lb. of steam—including moisture.

The figures in the third column give the number to be added to the value given in the second column for each one per cent. of moisture in the steam. Thus, for dry steam at 100lbs. absolute the value of m is 0·8773. For steam at the same pressure but with a dryness fraction of 80 per cent. (20 per cent. of moisture) the value of the index is increased to 0·8965.

In fact, if the percentage of moisture present is x we may write

$$m = 0{\cdot}8719 + 0{\cdot}000054\, p + x\,(0{\cdot}00106 - 0{\cdot}000001\, p).$$

Table LXI.—*Velocity at the neck of a nozzle.*

Initial Pressure, lbs. per sq. in.	Velocity in feet per second.			Initial pressure lbs. per sq. in.	Velocity in feet per second.		
	Dryness (Initial).				Dryness (Initial).		
	1·0	0·9	0·8		1·0	0·9	0·8
250	1497	1419	1336	110	1468	1390	1309
240	1496	1418	1335	100	1464	1386	1306
230	1495	1417	1334	90	1460	1382	1303
220	1494	1415	1333	80	1455	1377	1299
210	1493	1414	1332	70	1449	1372	1294
200	1491	1412	1330	60	1443	1366	1287
190	1489	1410	1328	50	1436	1358	1280
180	1487	1408	1326	40	1427	1349	1272
170	1485	1406	1324	30	1415	1339	1263
160	1483	1404	1322	20	1397	1322	1246
150	1480	1401	1320	15	1385	1311	1235
140	1477	1399	1318	10	1367	1295	1220
130	1474	1396	1315	5	1341	1268	1197
120	1471	1393	1312	1	1279	1210	1140

Explanation of the table :—

The velocity given in the table is that at the least section of a divergent nozzle expanding to a pressure of not more than 0·57 of the initial pressure approximately. All pressures are absolute. Both steam and moisture are moving with the same velocity.

The velocity is almost exactly proportional to the square root of the initial dryness fraction. What difference there is may be due to an error in determining m.

TABLE LXII.—*Discharge from Nozzles per Second.*

Initial Pressure, absolute.	Lbs. per sq. in. of least section; initial dryness.			Initial Pressure, absolute.	Lbs. per sq. in. of least section; initial dryness.		
	1·0	0·9	0·8		1·0	0·9	0·8
250	3·51	3·69	3·90	110	1·58	1·66	1·75
240	3·38	3·55	3·75	100	1·44	1·52	1·60
230	3·24	3·40	3·60	90	1·30	1·37	1·44
220	3·10	3·26	3·45	80	1·16	1·22	1·29
210	2·97	3·12	3·30	70	1·02	1·07	1·13
200	2·83	2·97	3·14	60	·880	·925	·978
190	2·69	2·83	2·99	50	·738	·777	·820
180	2·55	2·69	2·84	40	·595	·626	·661
170	2·42	2·54	2·69	30	·450	·474	·500
160	2·28	2·40	2·53	20	·303	·319	·337
150	2·14	2·25	2·38	15	·230	·242	·256
140	2·00	2·11	2·22	10	·1545	·1625	·172
130	1·86	1·96	2·07	5	·0791	·0833	·088
120	1·72	1·81	1·91	1	·0165	·0174	·0183

Explanation of the table :—

The table gives the weight of steam in lbs., including moisture — discharged from a nozzle having different initial pressures and dryness fractions, provided that the final pressure is not more than about 0·57 the initial pressure.

The weight discharged by a given nozzle is very nearly proportional to the absolute initial pressure, and almost exactly inversely proportional to the square root of the initial dryness. More nearly if the initial dryness is, say, 0·8 then the discharge can be calculated from the discharge of dry steam by merely dividing by 0·9. If the dryness is 0·9 the divisor becomes 0·95 ; and so on. The table illustrates the necessity of knowing the quality of the steam in nozzle experiments. Experiments by Prof. Rateau give discharges higher than our calculated values for dry steam by from one to two per cent. This is probably due to the presence of a slight amount of moisture. The above table does not apply to orifices in thin plates because the area of the orifice is greater than the least section of the jet.

Table LXIII.

n	0·105	1·115	1·125	1·135	1·145	1·155
m	·9051	·8971	·8890	·8810	·8734	·8658
$\left(\frac{u_0}{u_1}\right)$	1·630	1·628	1·626	1·624	1·622	1·620

Explanation of the table :—

The table gives the ratio of the volume per lb. for the steam at the neck and previous to entering the nozzle. This ratio is very nearly constant at 1·625. n and m are the indexes of the expansion curve as used throughout this book.

Table LXIV.

Showing the relation between the quantity of heat converted into kinetic energy and the velocity generated by this conversion, the initial velocity being zero. This table will be found useful when constructing velocity curves for the heat and work diagrams.

H = heat in B.Th.U.
v = velocity in feet per second.

H	v	H	v	H	v	H	v
1	224	7	592	25	1,119	100	2,238
2	317	8	633	30	1,225	150	2,740
3	387	9	671	40	1,414	200	3,165
4	447	10	707	50	1,580	250	3,535
5	500	15	866	60	1,732	300	3,870
6	548	20	1,000	80	2,000	350	4,180

$$v = 223{\cdot}8 \sqrt{H}.$$

The author has determined the theoretically available work in steam expanding adiabatically between different pressures. The results are given below in Table LXV. The work obtainable from the initially dry steam and the correction *to be deducted* for each one per cent. of moisture present are given for back pressures of 14·7—atmospheric—2, 1·5 and 1lb. absolute. For instance, if the steam is initially dry at 150lbs. absolute and expands to a back pressure of 1·5lbs. the theoretical work done

during adiabatic expansion is 293 B.Th.U. If the initial steam contained 2 per cent. of moisture we must deduct $2 \times 2 \cdot 55$, or 5·1 B.Th.U. from the above figure.

TABLE LXV.

Initial Pressure, lbs. abs.	Available Work, B.Th.U.				Correction per one per cent. of Moisture, in B.Th.U.			
	Back Pressure, lbs. per square inch absolute.				Back Pressure, lbs. square in.			
p	14·7	2	1·5	1	14·7	2	1·5	1
250	203	311	326	342	1·81	2·63	2·74	2·86
245	202	310	325	341	1·80	2·63	2·73	2·86
240	201	309	324	340	1·79	2·62	2·73	2·85
235	200	308	323	339	1·78	2·62	2·72	2·84
230	199	306	321	338	1·77	2·61	2·71	2·84
225	198	305	320	337	1·76	2·60	2·71	2·83
220	196	303	318	335	1 75	2·59	2·70	2·83
215	194	302	317	334	1·74	2·58	2·69	2·82
210	192	300	315	332	1·73	2·58	2·68	2·81
205	191	299	313	331	1·71	2·57	2·67	2·81
200	189	297	311	329	1·70	2·56	2·66	2·80
195	187	296	310	328	1·68	2·55	2·65	2·79
190	185	294	308	326	1·67	2·54	2·64	2·78
185	183	292	307	325	1·65	2·53	2·63	2·77
180	181	290	305	323	1·64	2·51	2·62	2·76
175	179	288	303	322	1·62	2·50	2·61	2·75
170	176	286	301	320	1·61	2·49	2·60	2·74
165	174	284	299	318	1·59	2·47	2·59	2·73
160	171	282	297	316	1·58	2·46	2·58	2·72
155	169	280	295	314	1·56	2·45	2·56	2·71
150	166	277	293	312	1·54	2·43	2·55	2·70
145	164	275	291	310	1·52	2·42	2·53	2 68
140	161	273	289	308	1·50	2·40	2·52	2·67
135	158	271	287	306	1·47	2·39	2·50	2·65
130	155	268	284	303	1·45	2·37	2·48	2·63
125	152	265	282	301	1·43	2·35	2·46	2·62
120	149	262	279	298	1·40	2·34	2·45	2·60
115	146	259	276	296	1·38	2·32	2·43	2·59
110	143	256	273	293	1·35	2·30	2·41	2·57
105	140	253	270	290	1·32	2·28	2·39	2·55
100	136	250	266	287	1·29	2·26	2·37	2·53
95	132	247	263	284	1·26	2·24	2·35	2·51
90	128	244	259	280	1·23	2·21	2·32	2·49

Initial Pressure, lbs. abs.	Available Work, B.Th.U. Back Pressure, lbs. per square inch absolute.				Correction per one per cent. of Moisture, in B.Th.U. Back Pressure, lbs. square inch.			
p	14·7	2	1·5	1	14·7	2	1·5	1
85	124	240	255	276	1·20	2·18	2·29	2·46
80	119	236	251	272	1·16	2·15	2·26	2·43
75	114	232	247	268	1·12	2·12	2·23	2·40
70	109	227	243	263	1·07	2·08	2·20	2·37
65	104	222	238	258	1·02	2·05	2·17	2·33
60	98	217	232	252	·97	2·01	2·13	2·29
55	93	212	227	247	·91	1·96	2·09	2·25
50	87	206	221	241	·85	1·91	2·04	2·21
45	79	199	214	234	·78	1·85	1·98	2·16
40	70	191	206	227	·70	1·79	1·92	2·11
35	61	182	198	219	·62	1·72	1·85	2·04
30	50	173	189	210	·50	1·63	1·77	1·96
25	37	161	179	200	·38	1·53	1·67	1·86
20	20	148	166	189	·22	1·40	1·55	1·74
19		145	163	186		1·37	1·52	1·71
18		141	160	183		1·34	1·49	1·68
17		137	157	179		1·31	1·46	1·65
16		134	153	176		1·28	1·43	1·62
15		130	149	172		1·25	1·40	1·58
14		125	144	168		1·21	1·36	1·55
13		121	139	163		1·16	1·32	1·51
12		116	134	158		1·12	1·27	1·47
11		111	128	152		1·07	1·22	1·42
10		106	122	145		1·00	1·16	1·36
9		100	115	139		·93	1·09	1·30
8		93	108	131		·86	1·02	1·24
7		86	99	123		·78	·94	1·16
6		78	90	114		·69	·85	1·07
5		70	80	104		·58	·74	·97

TABLE LXVI.

p	Q	p	Q	p	Q
1	·3170	29	·01479	58	·00796
1·5	·2175	30	·01435	59	·00784
2	·1675			60	·00772
3	·1155	31	·01393		
4	·0889	32	·01353	61	·00760
5	·0725	33	·01316	62	·00749
6	·0614	34	·01282	63	·00739
7	·0534	35	·01249	64	·00729
8	·0473	36	·01218	65	·00739
9	·0425	37	·01189	66	·00709
10	·0386	38	·01160	67	·00699
		39	·01133	68	·00690
11	·0354	40	·01108	69	·00681
12	·0328			70	·00673
13	·0305	41	·01084		
14	·0285	42	·01060	71	·00665
14·7	·0273	43	·01038	72	·00656
15	·0268	44	·01017	73	·00648
16	·0253	45	·00997	74	·00640
17	·0239	46	·00978	75	·00632
18	·0227	47	·00960	76	·00625
19	·02165	48	·00942	77	·00618
20	·02066	49	·00925	78	·00611
		50	·00908	79	·00604
21	·01976			80	·00597
22	·01896	51	·00892		
23	·01823	52	·00877	81	·00591
24	·01754	53	·00862	82	·00585
25	·01689	54	·00848	83	·00579
26	·01630	55	·00834	84	·00572
27	·01575	56	·00821	85	·00566
28	·01525	57	·00808	86	·00560

p	Q	p	Q	p	Q
87	·00554	117	·00427	147	·00348
88	·00549	118	·00423	148	·00346
89	·00544	119	·00420	149	·00344
90	·00539	120	·00417	150	·00342
91	·00533	121	·00413	151	·00340
92	·00528	122	·00410	152	·00338
93	·00523	123	·00408	153	·00336
94	·00518	124	·00405	154	·00335
95	·00513	125	·00402	155	·00333
96	·00508	126	·00399	156	·00331
97	·00504	127	·00396	157	·00329
98	·00500	128	·00394	158	·00327
99	·00495	129	·00391	159	·00325
100	·00490	130	·00388	160	·00324
101	·00486	131	·00386	161	·00322
102	·00482	132	·00383	162	·00320
103	·00478	133	·00381	163	·00318
104	·00474	134	·00378	164	·00317
105	·00470	135	·00375	165	·00315
106	·00465	136	·00373	166	·00313
107	·00461	137	·00371	167	·00312
108	·00458	138	·00368	168	·00310
109	·00454	139	·00366	169	·00308
110	·00451	140	·00364	170	·00306
111	·00447	141	·00361	171	·00305
112	·00443	142	·00359	172	·00303
113	·00440	143	·00357	173	·00301
114	·00436	144	·00355	174	·00300
115	·00433	145	·00353	175	·00298
116	·00430	146	·00350	176	·00297

p	Q	p	Q	p	Q
177	·00295	207	·00258	237	·00229
178	·00294	208	·00257	238	·00228
179	·00292	209	·00255	239	·00227
180	·00291	210	·00254	240	·00227
181	·00290	211	·00253	241	·00226
182	·00288	212	·00252	242	·00225
183	·00287	213	·00251	243	·00224
184	·00285	214	·00250	240	·00223
185	·00284	215	·00249	245	·00222
186	·00283	216	·00248	246	·00221
187	·00282	217	·00247	247	·00220
188	·00280	218	·00246	248	·00220
189	·00279	219	·00245	249	·00219
190	·00278	220	·00244	250	·00218
191	·00277	221	·00243		
192	·00276	222	·00242		
193	·00275	223	·00241		
194	·00273	224	·00240		
195	·00272	225	·00240		
196	·00271	226	·00239		
197	·00270	227	·00238		
198	·00269	228	·00237		
199	·00267	229	·00236		
200	·00266	230	·00235		
201	·00265	231	·00234		
202	·00264	232	·00233		
203	·00263	233	·00232		
204	·00262	234	·00231		
205	·00260	235	·00230		
206	·00259	236	·00230		

p = absolute pressure in lbs. per square inch.

Q = volume in cubic feet per B.Th.U. on the heat diagram, as described in Chapter XIII.

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
1	102·02	70·04	1043·02	1113·06	61·62	330·4	·00303
2	126·30	94·37	1026·09	1120·46	64·11	171·9	·00582
3	141·65	109·76	1015·38	1125·14	65·65	117·3	·00852
4	153·12	121·27	1007·37	1128·64	66·77	89·5	·01117
5	162·37	130·56	1000·90	1131·46	67·66	72·56	·01378
6	170·17	138·40	995·44	1133·84	68·40	61·14	·01636
7	176·94	145·21	990·69	1135·91	69·04	52·89	·01891
8	182·95	151·25	986·48	1137·74	69·60	46·65	·02144
9	188·36	156·70	982·69	1139·39	70·11	41·77	·02394
10	193·28	161·66	979·23	1140·89	70·56	37·83	·02644
11	197·81	166·23	976·05	1142·27	70·97	34·59	·02891
12	202·01	170·46	973·10	1143·55	71·33	31·87	·03138
13	205·93	174·40	970·35	1144·75	71·66	29·56	·03383
14	209·60	178·11	967·76	1145·87	71·97	27·58	·03626
14·7	212·00	180·53	966·07	1146·60	72·18	26·37	·03793
15	213·07	181·61	965·32	1146·93	72·27	25·85	·03869
16	216·35	184·92	963·01	1147·93	72·55	24·33	·04111
17	219·45	188·06	960·82	1148·87	72·81	22·98	·04352
18	222·42	191·06	958·72	1149·78	73·06	21·78	·04592
19	225·26	193·92	956·73	1150·64	73·30	20·70	·04831
20	227·96	196·66	954·81	1151·47	73·52	19·73	·05069
21	230·57	199·29	952·98	1152·26	73·74	18·84	·05307
22	233·07	201·82	951·21	1153·03	73·94	18·04	·05545
23	235·48	204·26	949·50	1153·76	74·14	17·30	·05781
24	237·80	206·61	947·86	1154·47	74·32	16·62	·06017
25	240·05	208·89	946·27	1155·16	74·50	16·00	·06252
26	242·23	211·09	944·73	1155·82	74·68	15·42	·06487
27	244·33	213·22	943·24	1156·46	74·85	14·88	·06721
28	246·38	215·29	941·79	1157·08	75·01	14·38	·06955
29	248·36	217·31	940·38	1157·69	75·17	13·91	·07187
30	250·29	219·26	939·02	1158·28	75·32	13·48	·07420

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
31	252·17	221·17	937·69	1158·85	75·47	13·07	·07652
32	254·00	223·02	936·39	1159·41	75·61	12·68	·07884
33	255·78	224·83	935·13	1159·95	75·74	12·32	·08115
34	257·52	226·59	933·89	1160·48	75·88	11·98	·08346
35	259·22	228·32	932·69	1161·00	76·01	11·66	·08577
36	260·88	230·00	931·51	1161·51	76·13	11·36	·08807
37	262·51	231·65	930·35	1162·00	76·26	11·07	·09036
38	264·09	233·26	929·23	1162·49	76·38	10·79	·09266
39	265·65	234·84	928·12	1162·96	76·49	10·53	·09495
40	267·17	236·39	927·04	1163·43	76·61	10·28	·09723
41	268·66	237·90	925·98	1163·88	76·72	10·05	·09951
42	270·12	239·39	924·94	1164·33	76·83	9·83	·10179
43	271·56	240·85	923·92	1164·77	76·93	9·61	·10407
44	272·97	242·28	922·92	1165·19	77·04	9·40	·10635
45	274·35	243·68	921·94	1165·62	77·14	9·21	·10862
46	275·70	245·06	920·97	1166·03	77·24	9·02	·11088
47	277·04	246·42	920·02	1166·44	77·33	8·84	·11315
48	278·35	247·75	919·08	1166·84	77·43	8·67	·11541
49	279·64	249·06	918·16	1167·23	77·52	8·50	·11767
50	280·90	250·36	917·26	1167·62	77·61	8·34	·11993
51	282·15	251·62	916·37	1167·99	77·69	8·19	·12218
52	283·38	252·87	915·49	1168·37	77 78	8·04	·12443
53	284·59	254·11	914·63	1168·74	77·87	7·89	·12668
54	285·78	255·32	913·78	1169·10	77·95	7·76	·12893
55	286·95	256·52	912·94	1169·46	78·04	7·62	·13117
56	288·11	257·69	912·12	1169·81	78·12	7·49	·13341
57	289·25	258·86	911·30	1170·16	35·20	7·37	·13565
58	290·37	260·00	910·50	1170·50	72·24	7·25	·13789
59	291·48	261·13	909·71	1170·84	77·87	7·14	·14013
60	292·57	262·25	908·93	1171·18	88 87	7·02	·14236

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
61	293·65	263·35	908·16	1171·51	78·49	6·92	·14459
62	294·72	264·43	907 39	1171·83	78·57	6·81	·14682
63	295·77	265·51	906·64	1172·15	78·64	6·71	·14905
64	296·81	266·57	905 90	1172·47	78·71	6·61	·15128
65	297·83	267 61	905·17	1172·78	78·78	6·52	·15350
66	298·84	268·64	904·44	1173·09	78·85	6·42	·15572
67	299·84	269·67	903·73	1173·39	78·91	6·33	·15794
68	300·83	270·67	903·02	1173·69	78·98	6·24	·16016
69	301·81	271·67	902 32	1173·99	79·04	6·16	·16237
70	302·77	272·66	901·63	1174·29	79·11	6·08	·16458
71	303·73	273·63	900·95	1174·58	79·17	5·99	·16679
72	304·67	274·60	900·27	1174·87	79·23	5·92	·16900
73	305·60	275·55	899·60	1175·15	79·29	5·84	·17121
74	306·53	276·49	898·94	1175·43	79 35	5·77	·17342
75	307·44	277·43	898·28	1175·71	79·41	5·69	·17562
76	308·34	278·35	897·64	1175·99	79·47	5·62	·17783
77	309·24	279 27	896·99	1176·26	79·53	5·56	·18003
78	310·12	280·17	896·36	1176·53	79·58	5·49	·18223
79	311·00	281·07	895·73	1176·80	79·64	5·42	·18443
80	311·87	281·95	895·11	1177·06	79·69	5·36	·18663
81	312·73	282·83	894·49	1177·32	79·75	5 30	·18882
82	313·58	283·70	893·88	1177·58	79·80	5·24	·19102
83	314·42	284·56	893·28	1177·84	79·86	5·18	·19321
84	315·25	285·41	892 68	1178·09	79·91	5·12	·19540
85	316·08	286·26	892·08	1178·34	79·96	5·06	·19759
86	316·89	287·10	891·50	1178·59	80·01	5·01	·19978
87	317·71	287·93	890·91	1178·84	80·06	4·95	·20197
88	318·51	288·75	890·34	1179·09	80·11	4·90	·20416
89	319·31	289·57	889·76	1179·33	80·16	4·85	·20634
90	320·09	290·37	889·20	1179·57	80·21	4·80	·20853

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to $t°$ F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at $t°$ into Steam at $t°$.	Total Heat of Evaporation from 32° F. and at $t°$ F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
91	320·88	291·18	888·63	1179·81	80·26	4·75	·21071
92	321·65	291·97	888·08	1180·05	80·31	4·70	·21289
93	322·42	292·76	887·52	1180·28	80·35	4·65	·21507
94	323·18	293·54	886 97	1180·51	80·40	4·60	·21725
95	323·94	294·31	886·43	1180·74	80·44	4·56	·21943
96	324·69	295·08	885·89	1180·97	80·49	4·51	·22160
97	325·43	295·85	885·35	1181 20	80·53	4·47	·22378
98	326·17	296·60	884·82	1181·42	80·58	4·43	·22595
99	326·90	297·35	884·30	1181·65	80·62	4·38	·22812
100	327·63	298·09	883·77	1181·87	80·67	4·34	·23029
101	328·35	298·83	883·25	1182·09	80·71	4·30	·23246
102	329·06	299·57	882·74	1182·30	80·75	4·26	·23463
103	329·77	300·29	882·23	1182·52	80·79	4·22	·23680
104	330·47	301·01	881·72	1182·73	80·84	4·19	·23897
105	331·17	301·73	881·21	1182·95	80·88	4·15	·24114
106	331·86	302·44	880·71	1183·16	80·92	4·11	·24330
107	332 55	303·15	880·21	1183·37	80 96	4·07	·24547
108	333 23	303·85	879·72	1183·57	80·99	4·04	·24763
109	333·91	304·55	879·23	1183·78	81·03	4·00	·24979
110	334·58	305·24	878·74	1183·99	81·07	3·97	·25195
111	335·25	305·93	878·26	1184·19	81·11	3·93	·25411
112	335·91	306·61	877·78	1184·39	81·15	3·90	·25626
113	336·57	307·29	877·31	1184·59	81·18	3 87	·25842
114	337·23	307·96	876·84	1184·79	81·22	3·84	·26058
115	337·87	308·62	876·37	1184·99	81·26	3 81	·26273
116	338 52	309·28	875·91	1185·19	81·29	3·78	·26489
117	339·16	309·94	875·44	1185·38	81·33	3·75	·26704
118	339·80	310·59	874·99	1185·58	81·37	3·72	·26919
119	340·43	311·24	874·53	1185·77	81·40	3·69	·27135
120	341·06	311·89	874·08	1185·96	81·44	3·66	·27350

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
121	341·68	312·52	873·63	1186·15	81·47	3·63	·27565
122	342·30	313·16	873·18	1186·34	81·51	3·60	·27780
123	342·92	313·79	872·73	1186·53	81·54	3·57	·27995
124	343·53	314·42	872·29	1186·71	81·58	3·55	·28210
125	344·14	315·05	871·85	1186·90	81·61	3·52	·28424
126	344·74	315·67	871·41	1187·08	81·65	3·49	·28639
127	345·34	316·29	870·98	1187·27	81·68	3·47	·28853
128	345·94	316·90	870·55	1187·45	81·71	3·44	·29068
129	346·53	317·51	870·12	1187·63	81·74	3·42	·29282
130	347·12	318·12	869·69	1187·81	81·77	3·39	·29496
131	347·71	318·73	869·26	1187·99	81·81	3·37	·29710
132	348·29	319·33	868·84	1188·17	81·84	3·34	·29924
133	348·87	319·92	868·42	1188·34	81·87	3·32	·30138
134	349·44	320·52	868·01	1188·52	81·90	3·29	·30352
135	350·02	321·11	867·59	1188·69	81·93	3·27	·30566
136	350·58	321·69	867·18	1188·87	81·96	3·25	·30780
137	351·15	322·27	866·77	1189·04	81·99	3·23	·30993
138	351·71	322·85	866·36	1189·21	82·02	3·20	·31207
139	352·27	323·43	865·96	1189·38	82·05	3·18	·31420
140	352·83	324·00	865·55	1189·56	82·08	3·16	·31634
141	353·38	324·57	865·15	1189·72	82·11	3·14	·31847
142	353·93	325·14	864·75	1189·89	82·14	3·12	·32060
143	354·48	325·71	864·35	1190·06	82·17	3·10	·32273
144	355·02	326·27	863·96	1190·23	82·19	3·08	·32487
145	355·56	326·82	863·57	1190·39	82·22	3·06	·32700
146	356·10	327·38	863·18	1190·55	82·25	3·04	·32913
147	356·64	327·93	862·79	1190·72	82·28	3·02	·33126
148	357·17	328·48	862·40	1190·88	82·30	3·00	·33339
149	357·70	329·02	862·02	1191·04	82·33	2·98	·33552
150	358·22	329·57	861·63	1191·20	82·36	2·96	·33764

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
151	358·62	330·12	861·23	1191·36	82·38	2·94	·33986
152	359·14	330·63	860·85	1191·51	82·41	2·93	·34199
153	359·66	331·18	860·47	1191·67	82·43	2·91	·34411
154	360·17	331·72	860·10	1191·82	82·46	2·89	·34624
155	360·68	332·24	859·72	1191·98	82·48	2·88	·34836
156	361 20	332·77	859·35	1192·13	82·51	2·86	·35049
157	361·70	333·30	858·98	1192·28	82·53	2·84	·35261
158	362·21	333·82	858·61	1192·44	82·56	2·83	·35474
159	362·70	334·34	858·24	1192·61	82·59	2·81	·35686
160	363·34	334·85	857·91	1192·76	82·62	2·79	·35889
161	363·63	335·38	857·54	1192·91	82·64	2·78	·36101
162	364·20	335 89	857·16	1193·06	82·66	2·76	·36313
163	364·69	336·40	856·80	1193·21	82·69	2·75	·36524
164	365·18	336·91	856·44	1193·36	82·71	2·73	·36736
165	365·68	337·41	856·09	1193·50	82·73	2·72	·36947
166	366·17	337·92	855·71	1193·65	82·75	2·70	·37160
167	366·65	338·41	855·36	1193·80	82·78	2·68	·37372
168	367·13	338·90	855·00	1193·95	82·80	2·67	·37583
169	367·62	339·41	854·66	1194·10	82·82	2·65	·37795
170	368·23	339·89	854·36	1194·25	82·85	2·63	·38007
171	368·59	340·38	853·95	1194·39	82·87	2·62	·38219
172	369·04	340·87	853·61	1194·53	82·89	2·60	·38430
173	369·51	341·35	853·28	1194·67	82·92	2·59	·38641
174	369·98	341·84	852·13	1194·82	82·94	2·57	·38853
175	370·45	342·33	852·60	1194·96	82·96	2·56	·39064
176	370·90	342·80	852·27	1195·10	82 98	2·54	·39274
177	371·37	343·29	851·93	1195·24	83·00	2·53	·39486
178	371·83	343·76	851·60	1195·38	83·03	2·51	·39697
179	372·29	344·23	851·27	1195·52	83·05	2·50	·39908
180	372·89	344·71	850·96	1195·67	83·07	2·49	·40120

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
181	373·20	345·18	850·54	1195·80	83·09	2·48	·40331
182	373·66	345·65	850·22	1195·94	83·11	2·47	·40542
183	374·11	346·12	849·89	1196·08	83·13	2·45	·40754
184	374·56	346·58	849·57	1196·22	83·15	2·44	·40965
185	375·01	347·05	849·26	1196·36	83·17	2·43	·41175
186	375·45	347·51	848·95	1196·48	83·19	2·42	·41386
187	375·90	347·97	848·63	1196·62	83·21	2·40	·41596
188	376·34	348·42	848·30	1196·76	83·23	2·39	·41807
189	376·78	348·87	847·99	1196·90	83·25	2·38	·42020
190	377·35	349·33	847·70	1197·03	83·27	2·37	·42228
191	377·22	349·77	847·38	1197·16	83·29	2·36	·42438
192	378·09	350·22	847·06	1197·29	83·31	2·35	·42648
193	378·52	350·67	846·75	1197·42	83·32	2·34	·42859
194	378·95	351·12	846·43	1197·55	83·34	2·33	·43069
195	379·38	351·57	846·12	1197·68	83·35	2·31	·43279
196	379·90	352·02	845·80	1197·81	83·37	2·30	·43490
197	380·23	352·45	845·50	1197·94	83·39	2·29	·43700
198	380·65	352·89	845·19	1198·07	83·41	2·28	·43910
199	381·08	353·33	844·87	1198·20	83·43	2·27	·44121
200	381·64	353·77	844·57	1198·34	83·46	2·26	·44331
201	382·1	354·1	844·3	1198·46	83·48	2·250	·4453
202	382·5	354·6	844·0	1198·59	83·50	2·236	·4475
203	382·9	355	843·7	1198·71	83·51	2·227	·4496
204	383·3	355·3	843·3	1198·83	83·53	2·217	·4516
205	383·7	355·8	843·0	1198·96	83·55	2·206	·4538
206	384·1	356·3	842·7	1199·09	83·57	2·196	·4558
207	384·5	356·8	842·4	1199·21	83·59	2·186	·4580
208	384·9	357·2	842·1	1199·33	83·61	2·176	·4600
209	385·3	357·7	841·8	1199·46	83·63	2·166	·4621
210	385·7	358·1	841·5	1199·57	83·65	2·155	·4644

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
211	386·1	358·6	841·2	1199·71	83·66	2·146	·4664
212	386·5	359	840·9	1199·83	83·68	2·137	·4684
213	386·9	359·4	840·6	1199·95	83·70	2·128	·4706
214	387·3	359·9	840·3	1200·08	83·72	2·119	·4726
215	387·7	360·2	840·0	1200·20	83·73	2·109	·4746
216	388·0	360·6	839·8	1200·32	83·75	2·100	·4766
217	388·4	361·1	839·5	1200·45	83·77	2·090	·4787
218	388·8	361·5	839·2	1200·57	83·78	2·081	·4808
219	389·3	361·9	838·9	1200·69	83·80	2·072	·4830
220	389·7	362·3	838·6	1200·82	83·82	2·062	·4850
221	390·1	362·7	838·3	1200·95	83·84	2·053	·4870
222	390·5	363·1	838·0	1201·07	83·85	2·045	·4891
223	390·9	363·5	837·8	1201·19	83·87	2·036	·4912
224	391·3	363·9	837·5	1201·30	83·88	2·028	·4934
225	391·6	364·3	837·2	1201·43	83·89	2·020	·4956
226	392·0	364·8	836·9	1201·55	83·91	2·011	·4977
227	392·4	365·1	836·6	1201·66	83·93	2·002	·4999
228	392·8	365·5	836·3	1201·77	83·94	1·994	·5020
229	393·2	365·9	836·0	1201·89	83·96	1·985	·5040
230	393·6	366·3	835·8	1202·01	83·97	1·976	·5062
231	394·0	366·7	835·5	1202·11	83·99	1·968	·5082
232	394·3	367·1	835·3	1202·22	84·00	1·960	·5103
233	394·7	367·5	835·0	1202·33	84·01	1·952	·5124
234	395·1	367·9	834·8	1202·44	84·03	1·944	·5145
235	395·5	368·2	834·5	1202·55	84·04	1·936	·5166
236	395·9	368·6	834·3	1202·66	84·06	1·928	·5186
237	396·3	369	834·0	1202·77	84·08	1·920	·5208
238	396·6	369·4	833·7	1202·88	84·09	1·910	·5228
239	397·0	369·8	833·4	1202·99	84·10	1·904	·5249
240	397·4	370·1	833·1	1203·10	84·12	1·897	·5270

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
241	397·8	370·5	832·8	1203·21	84·13	1·889	·5290
242	398·1	370·9	832·6	1203·32	84·15	1·882	·5310
243	398·5	371·3	832·3	1203·43	84·17	1·875	·5330
244	398·9	371·7	832·1	1203·53	84·18	1·868	·5351
245	399·2	372·1	831·8	1203·66	84·20	1·861	·5372
246	399·6	372·4	831·5	1203·77	84·21	1·853	·5393
247	400·0	372·8	831·3	1203·88	84·22	1·847	·5414
248	400·3	373·2	831·0	1203·00	84·24	1·839	·5435
249	400·7	373·6	830·7	1204·10	84·25	1·832	·5455
250	401·1	374·0	830·5	1204·21	84·27	1·825	·5476
251	401·4	374·3	830·2	1204·32	84·29	1·818	·5497
252	401·7	374·7	830·0	1204·43	84·30	1·812	·5517
253	402·1	375·0	829·8	1204·54	84·31	1·805	·5538
254	402·4	375·3	829·5	1204·65	84·32	1·798	·5559
255	402·7	375·7	829·2	1204·75	84·33	1·792	·5580
256	403·1	376·0	828·9	1204·86	84·34	1·785	·5601
257	403·5	376·3	828·6	1204·97	84·35	1·778	·5622
258	403·9	376·7	828·3	1205·08	84·36	1·772	·5643
259	404·2	377·1	828·0	1205·19	84·38	1·765	·5664
260	404·6	377·4	827·8	1205·29	84·39	1·759	·5685
261	404·9	377·8	827·6	1205·39	84·40	1·752	·5705
262	405·2	378·2	827·3	1205·50	84·42	1·747	·5725
263	405·5	378·5	827·1	1205·60	84·43	1·740	·5746
264	405·8	378·8	826·8	1205·70	84·44	1·733	·5767
265	406·1	379·2	826·6	1205·80	84·45	1·728	·5788
266	406·5	379·6	826·3	1205·90	84·47	1·721	·5810
267	406·9	379·9	826·0	1206·00	84·48	1·716	·5830
268	407·2	380·2	825·7	1206·11	84·49	1·710	·5851
269	407·5	380·6	825·5	1206·21	84·50	1·703	·5872
270	407·8	381·0	825·3	1206·31	84·51	1·697	·5894

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in Pounds per square inch = Gauge press. + 15.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.*, B. T. U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.*, B. T. U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B. T. U.	Volume of 1 lb. Steam in cubic feet $v = u + \sigma$.	Weight of 1 cubic foot of Steam in pounds.
p	t	S	L	H	$\frac{Pu}{J}$	v	w
271	408·1	381·3	825·0	1206·41	84·52	1·691	·5915
272	408·4	381·7	824·8	1206·51	84·54	1·685	·5935
273	408·8	382·0	824·6	1206·61	84·55	1·680	·5956
274	409·1	382·3	824·3	1206·71	84·56	1·674	·5978
275	409·4	382·6	824·0	1206·81	84·58	1·668	·5999
276	409·8	383·0	823·8	1206·91	84·59	1·663	·6020
277	410·0	383·3	823·6	1207·02	84·60	1·657	·6040
278	410·4	383·6	823·4	1207·12	84·61	1·650	·4060
279	410·8	384·0	823·1	1207·22	84·62	1·645	·6080
280	411·1	384·3	822·9	1207·32	84·63	1·639	·6101
281	411·4	384·7	822·7	1207·41	84·64	1·633	·6122
282	411·8	385·0	822·5	1207·51	84·66	1·628	·6143
283	412·1	385·3	822·2	1207·61	84·67	1·623	·6164
284	412·4	385·6	822·0	1207·70	84·68	1·617	·6185
285	412·7	386·0	821·7	1207·80	84·69	1·612	·6206
286	413·0	386·3	821·5	1207·90	84·70	1·606	·6227
287	413·4	386·6	821·3	1208·00	84·71	1·601	·6248
288	413·7	387·0	821·0	1208·10	84·72	1·596	·6269
289	414·0	387·3	820·8	1208·20	84·73	1·590	·6290
290	414·3	387·6	820·6	1208·30	84·74	1·585	·6310
291	414·6	387·9	820·3	1208·40	84·75	1·580	·6330
292	415·0	388·2	820·1	1208·49	84·76	1·575	·6351
293	415·3	388·5	819·9	1208·59	84·77	1·570	·6371
294	415·6	388·8	819·7	1208·68	84·78	1·565	·6392
295	415·9	389·2	819·5	1208·77	84·79	1·560	·6414
296	416·2	389·5	819·2	1208·86	84·80	1·555	·6434
297	416·6	389·8	819·0	1208·95	84·81	1·550	·6455
298	416·9	390·1	818·8	1209·06	84·82	1·545	·6475
299	417·2	390·4	818·6	1209·15	84·83	1·540	·6495
300	417·5	390·8	818·4	1209·25	84·84	1·536	·6515

TEMPERATURE PRESSURE TABLE.

Temp. Fah.	Absolute Pressure in lbs. per sq. in.	Temp. Fah.	Absolute Pressure in lbs. per sq. in.	Temp. Fah.	Absolute Pressure in lbs. per sq. in.	Temp. Fah.	Absolute Pressure in lbs. per sq. in.	Temp. Fah.	Absolute Pressure in lbs. per sq. in.
60	·26	100	·94	140	2·88	180	7·5	220	17·2
61	·26	101	·97	141	2·95	181	7·7	221	17·5
62	·27	102	1·00	142	3·03	182	7·8	222	17·9
63	·28	103	1·03	143	3·11	183	8·0	223	18·2
64	·29	104	1·06	144	3·19	184	8·2	224	18·6
65	·30	105	1·09	145	3·27	185	8·4	225	18·9
66	·31	106	1·13	146	3·35	186	8·6	226	19·3
67	·32	107	1·16	147	3·44	187	8·8	227	19·7
68	·33	108	1·19	148	3·53	188	8·9	228	20·0
69	·35	109	1·23	149	3·62	189	9·1	229	20·4
70	·36	110	1·27	150	3·7	190	9·3	230	20·8
71	·37	111	1·30	151	3·8	191	9·5	231	21·2
72	·38	112	1·34	152	3·9	192	9·7	232	21·6
73	·40	113	1·38	153	4·0	193	10·0	233	22·0
74	·41	114	1·42	154	4·1	194	10·2	234	22·4
75	·42	115	1·46	155	4·2	195	10·4	235	22·8
76	·44	116	1·50	156	4·3	196	10·6	236	23·3
77	·45	117	1·55	157	4·4	197	10·8	237	23·7
78	·47	118	1·59	158	4·5	198	11·1	238	24·1
79	·49	119	1·64	159	4·6	199	11·3	239	24·6
80	·50	120	1·68	160	4·7	200	11·5	240	25·0
81	·52	121	1·73	161	4·8	201	11·8	241	25·5
82	·53	122	1·78	162	5·0	202	12·0	242	25·9
83	·55	123	1·83	163	5·1	203	12·3	243	26·4
84	·57	124	1·88	164	5·2	204	12·5	244	26·9
85	·59	125	1·93	165	5·3	205	12·8	245	27·4
86	·61	126	1·98	166	5·5	206	13·0	246	27·8
87	·63	127	2·04	167	5·6	207	13·3	247	28·3
88	·65	128	2·10	168	5·7	208	13·6	248	28·9
89	·67	129	2·15	169	5·9	209	13·8	249	29·4
90	·69	130	2·21	170	6·0	210	14·1	250	29·9
91	·71	131	2·27	171	6·1	211	14·4	251	30·4
92	·74	132	2·33	172	6·3	212	14·7	252	30·9
93	·76	133	2·40	173	6·4	213	15·0	253	31·5
94	·78	134	2·46	174	6·6	214	15·3	254	32·0
95	·81	135	2·52	175	6·7	215	15·6	255	32·6
96	·83	136	2·59	176	6·9	216	15·9	256	33·2
97	·86	137	2·66	177	7·0	217	16·2	257	33·7
98	·89	138	2·73	178	7·2	218	16·5	258	34·3
99	·91	139	2·80	179	7·3	219	16·9	259	34·9

TEMPERATURE PRESSURE TABLE.

Temp. Fah.	Absolute Pressure in lbs. per sq. in.	Temp. Fah.	Absolute Pressure in lbs. per sq. in.	Temp. Fah.	Absolute Pressure in lbs. per sq. in.	Temp. Fah.	Absolute Pressure in lbs. per sq. in.	Temp. Fah.	Absolute Pressure in lbs. per sq. in.
260	35·5	300	67·2	340	118·4	380	196·3	420	308·9
261	36·1	301	68·2	341	120·0	381	198·7	421	312·3
262	36·7	302	69·3	342	121·6	382	201·1	422	315·6
263	37·4	303	70·3	343	123·3	383	203·5	423	319·0
264	38·0	304	71·4	344	124·9	384	205·9	424	322·4
265	38·6	305	72·4	345	126·6	385	208·3	425	325·9
266	39·3	306	73·5	346	128·2	386	210·8	426	329·3
267	39·9	307	74·6	347	129·3	387	213·3	427	332·8
268	40·6	308	75·7	348	131·6	388	215·8	428	336·4
269	41·3	309	76·8	349	133·4	389	218·3	429	339·9
270	42·0	310	77·9	350	135·1	390	220·9	430	343·5
271	42·7	311	79·1	351	136·9	391	223·5	431	347·1
272	43·4	312	80·2	352	138·7	392	226·1	432	350·7
273	44·1	313	81·4	353	140·5	393	228·7	433	354·4
274	44·8	314	82·6	354	142·3	394	231·4	434	358·0
275	45·5	315	83·8	355	144·1	395	234·0	435	361·7
276	46·3	316	85·0	356	146·0	396	236·7	436	365·5
277	47·0	317	86·2	357	147·8	397	239·4	437	369·2
278	47·8	318	87·5	358	149·7	398	242·2	438	373·0
279	48·6	319	88·7	359	151·6	399	245·0	439	376·8
280	49·3	320	90·0	360	153·6	400	247·8	440	380·7
281	50·1	321	91·3	361	155·5	401	250·6	441	384·6
282	50·9	322	92·5	362	157·5	402	253·4	442	388·5
283	51·7	323	93·9	363	159·5	403	256·3	443	392·4
284	52·6	324	95·2	364	161·5	404	259·2	444	396·4
285	53·4	325	96·5	365	163·5	405	262·1	445	400·4
286	54·2	326	97·9	366	165·5	406	265·0	446	404·4
287	55·1	327	99·2	367	167·6	407	268·0	447	408·4
288	56·0	328	100·6	368	169·7	408	271·0	448	412·5
289	56·8	329	102·0	369	171·8	409	274·0	449	416·6
290	57·7	330	103·4	370	173·9	410	277·1	450	420·7
291	58·6	331	104·9	371	176·1	411	280·1		
292	59·5	332	106·3	372	178·2	412	283·2		
293	60·5	333	107·8	373	180·4	413	286·3		
294	61·4	334	109·3	374	182·6	414	289·5		
295	62·3	335	110·7	375	184·9	415	292·7		
296	63·3	336	112·2	376	187·1	416	295·9		
297	64·3	337	113·8	377	189·4	417	299·1		
298	65·2	338	115·3	378	191·7	418	302·3		
299	66·2	339	116·9	379	194·0	419	305·6		

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